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ANALYSIS OF STEAM AND HYDRONIC COMPARTMENT HEATING SYSTEMS ABOARD U.S. COAST GUARD 140 FOOT WTGB CLASS CUTTERS

by

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ANALYSIS OF STEAM AND HYDRONIC COMPARTMENT HEATING SYSTEMS ABOARD U.S. COAST GUARD 140 FOOT WTGB CLASS CUTTERS

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ABSTRACT

The compartment heating system on the U.S. Coast Guard's Icebreaking Tug (WTGB) class cutter was studied to determine heat transfer performance characteristics of existing heat exchangers when used with circulating hot water vice steam. Characterizations such as Reynolds number vs. Colburn *j* factor plots, convection coefficients, overall coefficients, and Effectiveness-NTU relations were generated. Initial analysis with acknowledged conservative definitions of air side convection coefficients determined that the hydronic system provided on average seventy percent of the heat transfer capabilities available with the steam system. Improvements to the hydronic system were shown to increase heat exchanger performance parameters by an average of ten percent. It was notable that the added heat transfer available from steam is not due to a property of steam itself such as latent phase change effects, but is due solely to the increase in entering tube side temperature. Judging by heat transfer capabilities alone, with the described conservative assumptions on which these results are based, use of currently installed heat exchangers in a hydronic system is a viable option.

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LIST OF SYMBOLS

A_c	=	Cross sectional area
A_{ff}	=	Heat exchanger free flow area
A_{fr}	=	Heat exchanger frontal area
A_s	=	Surface area*
c_p	=	Constant pressure specific heat
C	=	Heat capacity rate
C_f	=	Friction coefficient
C_r	=	Heat capacity ratio
d_i	=	Tube inner diameter
d_o	=	Tube outer diameter
D_h	=	Flow passage hydraulic diameter
f	=	Friction factor
G	=	Mass velocity
h	=	Convection heat transfer coefficient
h_{fg}	=	Enthalpy difference due to condensation
j_H	=	Colburn j factor
k	=	Thermal conductivity
L	=	Length in direction of flow
L_{fin}	=	Fin length
L_{fin_c}	=	Fin corrected length
$\dot{m}$	=	Mass flow rate
NTU	=	Number of transfer units
Nu	=	Nusselt number
P	=	Fin perimeter
Pr	=	Prandtl number
$q_{\max}$	=	Fin maximum possible heat transfer rate
q_t	=	Fin total heat transfer rate
$\dot{Q}$	=	Heat transfer rate
Re	=	Reynolds number

LIST OF SYMBOLS (continued)

R_f = Fouling factor
R_t = Thermal resistance
R_{wall} = Tube wall conduction resistance
St = Stanton number
t = Specified component thickness
T_1 = Air side inlet temperature
T_2 = Air side outlet temperature
T_3 = Tube side inlet temperature
T_4 = Tube side outlet temperature
T_∞ = Ambient air temperature
T_{base} = Fin base temperature
T_M = Mean air temperature
T_S = Heat exchanger surface temperature
ΔT = Change in temperature
U = Overall heat transfer coefficient
V = Velocity
w_{fin} = Fin width

α = Heat exchanger area density
ϵ = Effectiveness
η_f = Single fin efficiency
η_o = Overall fin efficiency
μ = Dynamic viscosity
ν = Kinematic viscosity
ρ = Density

I. INTRODUCTION

A. CUTTER OPERATIONS

The United States Coast Guard operates nine Icebreaking Tug (WTGB) class cutters. Stationed in the northeast United States and on the Great Lakes, these cutters are primarily used for domestic icebreaking, but also routinely perform other missions such as search and rescue, pollution response, law enforcement, and aids to navigation support. The cutters are 140 feet long, have a beam of 37.5 feet, displace 662 tons, and are typically crewed by 17 personnel (see Appendix A for cutter illustration). The twin diesel-electric propulsion plant has a cruising range of 4,000 miles, maximum speed of 14.7 knots, and provides sufficient power for the hull to break through 18 to 20 inches of ice. A very capable and versatile platform, these cutters and crews make vital contributions to the overall service the Coast Guard provides to the public.

B. COMPARTMENT HEATING SYSTEM

1. Purpose

The primary purpose of the compartment heating system is to uphold the cutter's overall mission readiness, keeping on board personnel physically fit and mentally alert by providing an atmosphere of suitable air for breathing under conditions that will enable the body to maintain a proper heat balance. A secondary purpose of the system is provide suitable temperature and humidity conditions for the preservation of stores and equipment. [Ref. 1]

2. Description of Present Configuration

Compartment heating is accomplished either by duct heaters in the supply air ventilation system or by unit heaters mounted in the compartments themselves. These duct and unit heaters are either steam or electric. The scope of this thesis includes study of the steam duct and unit heaters only.

Heating of ventilation supply air is accomplished in two stages; first by a preheater and then by a reheater. Preheaters are usually located near supply air ventilation inlets, and heat incoming air sufficiently to prevent condensation in ventilation ducts. Reheaters are located in the compartments being heated, and further heat the air to a set room temperature. Unit heaters are installed in compartments which require a spot source of heat. [Ref. 1]

3. Deficiencies of Present Configuration

The present steam system in recent years has become very maintenance intensive, particularly with the auxiliary boilers. The current boilers are over 15 years old, are "water tube" type, and are difficult and expensive to procure spare parts for. With many cutters operating in demanding harsh winter conditions, reduction of heating equipment down time is a priority. The steam heating system equipment is also non-standard compared to other ship classes in the fleet, leading to support and maintenance difficulties.

4. Considerations for a Replacement System

The search for a suitable replacement for the boilers has led to consideration of a hydronic (circulating water in a closed piping system) compartment heating system to replace the current steam system. Major factors in choosing a replacement system are ease of installation and affordability. A way to ease the installation and decrease the cost of a new system is to leave in place and utilize as much of the equipment from the old system as possible. There is therefore considerable motivation to determine the heat transfer characteristics of the presently installed heat exchangers when used with circulating hot water.

II. HEAT EXCHANGER DESCRIPTIONS

A. SYSTEM OVERVIEW

The compartment heating system consists of a steam generation plant and various heat exchangers. Included in the steam generation plant are two auxiliary boilers, feedwater, chemical, and condensate tanks, and associated pumps, controls, piping, and valves. The boilers are rated at 620 pounds of saturated steam per hour at a working pressure of 35 psi with feedwater entering at 210 degrees F. A functional diagram of the steam heating system is shown in Figure 1.

B. HEAT EXCHANGERS

1. General

The heat exchangers installed on board the WTGB cutter class fall into two categories: unit heaters and duct heaters. Unit heaters consist of a coil, fan, motor, and casing assembly. They are installed in machinery and work spaces throughout the cutter. Duct heaters, as implied, are coils installed within the supply ventilation ducting, with separate ventilation fans installed upstream of coils. Duct heaters provide heat to the cutter's living spaces.

2. Unit Heaters

a. Dimensions

Unit heaters are manufactured by the New York Blower Company of Willowbrook, Illinois. Two models are employed: Model B-25 (quantity two) and Model

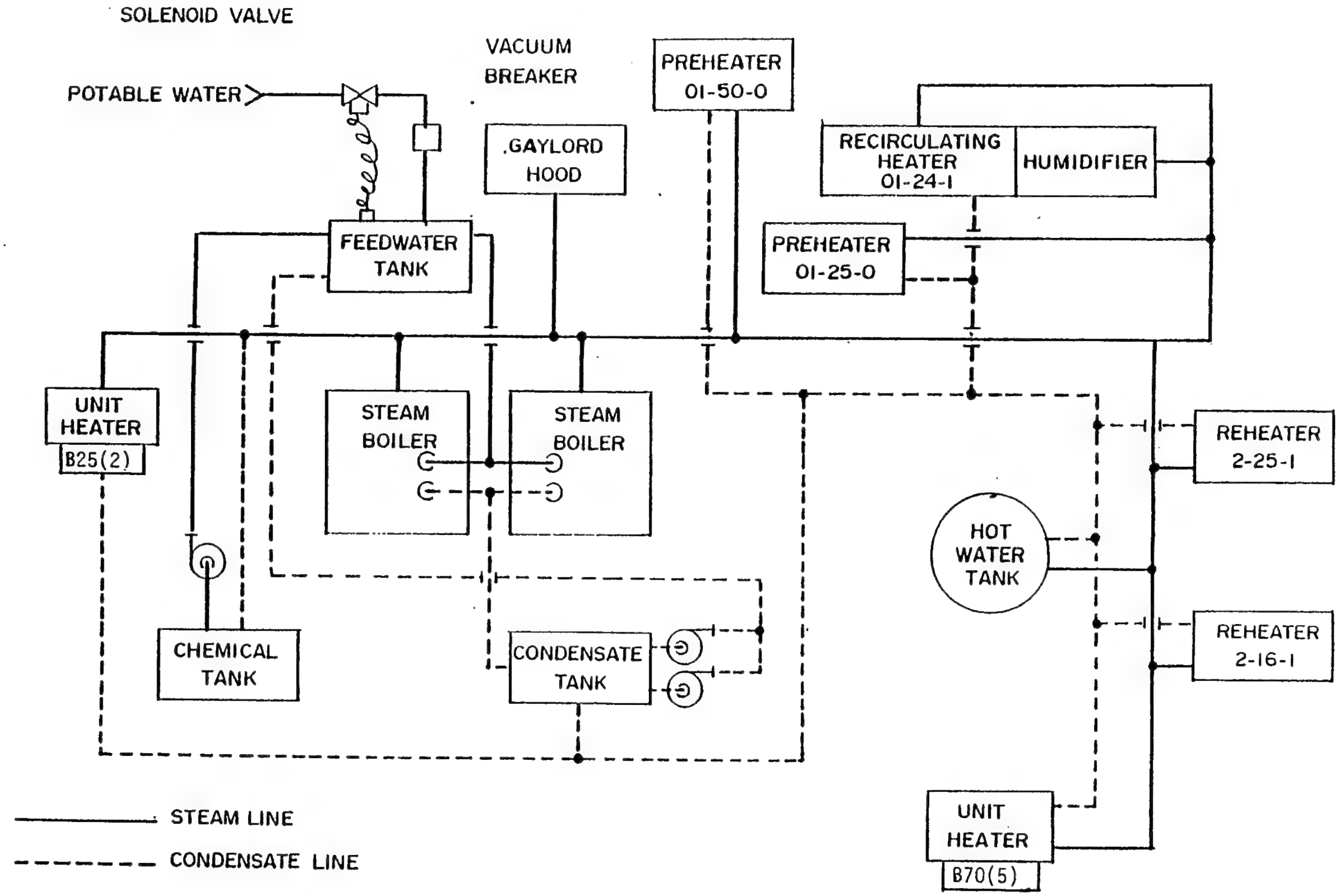


Figure 1. Steam Heating System Functional Diagram [Ref. 1]

B-70 (quantity five). Dimensions and material properties of unit heater coils and assemblies are shown in Appendix B. [Refs. 2-3]

b. Heating Capacities

Unit heaters are sized to operate with steam at 240 degrees F and 25 psig. For each unit heater, the location, air flow rate, and heat transfer rate stated on the ship's prints are shown in Table 1. [Refs. 4-7]

Heater	Location	Type	Air Flow (CFM)	Heat Transfer Stated on Prints (BTU/hr)
2-8-0	Paint Locker	B-70	1,505	5,294
2-40-1	Engine Room	B-70	1,505	64,860
2-40-2	Engine Room	B-70	1,505	64,860
2-60-1	Engine Room	B-70	1,505	64,860
2-60-2	Engine Room	B-70	1,505	64,860
2-73-1	Motor Room	B-25	400	33,120
2-80-1	Steering Gear	B-25	400	7,747

Table 1. Unit Heater Operating Parameters

3. Duct Heaters

a. Dimensions

Duct heaters are manufactured by two manufacturers: Colmac Coil Manufacturing Incorporated of Colville, Washington (quantity four) and Carrier

Corporation of Farmington, Connecticut (quantity one). Duct heater dimensions and material properties are shown in Appendix B. [Refs. 8-9]

b. Heating Capacities

Duct heaters are also sized to operate with steam at 240 degrees F and 25 psig. For each duct heater, the supply ventilation system, location, air flow rate, heat transfer rate, and air temperature rise stated on the ship's prints are shown in Table 2. [Refs. 4-7]

Heater	System	Location	Air Flow (CFM)	Heat Transfer Stated on Prints (BTU/hr)	Air Temp. Rise Stated on Prints (°F)
01-24-1	S 01-24-1	Fan Room	3,680	51,600	62 to 75
01-50-0	S 01-49-0	Exhst. Uptake	750	60,750	-30 to 45
01-25-1	S 01-27-1	Fan Room	1,400	113,400	-30 to 45
2-25-1	S 01-27-1	Aux. Mach. 1	1,050	45,360	40 to 80
2-16-1	S 01-27-1	Anchor Gear	350	20,034	45 to 98

Table 2. Duct Heater Operating Parameters

C. HEAT EXCHANGER DESIGN CONSIDERATIONS

1. Conventional Heat Exchangers

The design of a heat exchanger involves consideration of both the heat transfer rates between the fluids and the mechanical pumping power expended to overcome fluid friction. First examining heat transfer rates, there is a marked difference in heat exchanger performance depending on whether the fluids involved are gases or liquids.

The *convection heat transfer coefficient, h*, for gases is generally one or two orders of magnitude less than that for liquids. Correspondingly a gas heat exchanger's *convective thermal resistance, R_t*, defined as the inverse of the product of *h* and *heat exchanger surface area, A_s*, (i.e. $R_t = 1/hA_s$) is one or two orders of magnitude greater than that for a liquid heat exchanger. It is therefore evident that for the heat transfer rate of a gas heat exchanger to be equivalent to that of a liquid heat exchanger, the heat transfer surface area for a gas heat exchanger needs to be much larger than the heat transfer surface area for a liquid heat exchanger. [Refs. 10-11]

Examining the power required to overcome fluid - heat exchanger friction, conventional (e.g. concentric tube or shell and tube) heat exchangers operating with high density fluids have lesser frictional losses compared to heat exchangers operating with low density fluids. The pumping power required to move high density fluid (e.g. liquid) over a given heat exchanger is considerably less than the pumping power required to move low density fluid over the same heat exchanger. There can therefore be an equipment operating cost benefit to using a liquid heat exchanger system rather than a gas heat exchanger system.

Heat exchangers where at least one fluid is required to be a gas therefore need an improved design to make up for inherent drawbacks. The heat transfer obtained per unit of heat exchanger surface area can be increased by increasing the fluid flow velocity. This however is not a desirable method to increase heat transfer since the friction power expended to increase fluid flow velocity increases by as much as the cube of velocity. Minimizing friction power leads to limiting flow velocities, and this combined with the

relatively low thermal conductivity of most gases, results in low heat transfer rate per unit of heat exchanger surface area. There is therefore, in a conventional heat exchanger using gases, poor heat transfer performance at low flow velocities and an uneconomical, less than acceptable trade-off for increasing heat transfer per unit surface area by increasing flow velocities.

2. Compact Heat Exchangers

The development of compact heat exchangers is in response to the need to attain higher heat transfer rates with minimum space and power requirements. Large but compact surface areas are a typical characteristic of gas heat exchangers. Heat exchangers used on the WTGB cutter class heating system are one style of compact heat exchanger that incorporates large gas-side surface areas with dense arrays of continuous fins.

It is first noted that compactness itself leads to high performance. A compact surface has small flow passages and the heat transfer coefficient, h, varies as a negative power of the flow passage size. A customary expression for the size of a non-circular flow passage is the *hydraulic diameter,* D_h, equaling four times the cross-sectional area divided by the wetted perimeter. It is also true that a smaller hydraulic diameter increases friction power, but the benefits that compactness has on the heat transfer coefficient generally outweigh the detrimental influence of small hydraulic diameter on friction power. [Ref. 10]

In addition to the influence of small hydraulic diameter, increases in heat exchanger performance can be obtained by any modification of the surface geometry that

results in a higher heat transfer coefficient at a given flow velocity. One widely accepted modification is use of extended surfaces or fins so that in addition to providing increased heat transfer surface area, the interrupted surface prevents thickening boundary layers from reducing heat transfer. Finned surfaces also increase friction power and thermal resistance due to conduction, but a small improvement in the heat transfer coefficient can more than offset these negative factors. [Ref. 10]

Other methods of obtaining increased performance by change of flow surface geometry include the use of curved, corrugated, or wavy passages, in which boundary layer separation and turbulence (promoters of heat transfer) are induced. Such surfaces are incorporated in the duct heaters, but not in the unit heaters being studied.

A common descriptor of compact heat exchangers is the *area density,* α, which is the ratio of heat transfer surface area to heat exchanger volume. A conventional cutoff for labeling a heat exchanger as compact is an area density value greater than 700 square meters per cubic meter (or 213 square feet per cubic feet). This is not a staunch rule however, as many heat exchangers have been grouped into the compact category with lesser area densities [Ref. 10].

Compact heat exchanger designs provide the benefits of high heat transfer rate with minimum volume and thus are very well suited for duct heater applications. In shipboard systems where volume savings are invariably sought, use of compact heat exchangers is very common.

III. HEAT EXCHANGER ANALYSIS

A. STANDARD PRESENTATION OF PERFORMANCE DATA AND THE REYNOLDS ANALOGY

In engineering practice, it is often desirable to use a common presentation of performance data so as to avoid confusion associated with many arbitrarily defined parameters. In the study of heat transfer and flow-friction, commonality in data presentation is found using the *Reynolds Analogy*. As presented by Incropera and Dewitt [Ref. 11], and also Kays and Crawford [Ref. 12], with certain restrictions (noted shortly), relations that govern velocity boundary layer behavior are the same as those that govern the thermal boundary layer. From this it is known that non-dimensional friction and heat transfer relations for a particular geometry are closely related. Specifically, the *Reynolds number, Re*, the velocity boundary layer's *friction coefficient, C_f*, and the heat transfer boundary layer's *Nusselt number, Nu*, are related as follows:

$$C_f \frac{Re_L}{2} = Nu_L \qquad (1) \qquad\qquad where: \; Re_L = \frac{\rho V L}{\mu} \qquad (2)$$

$$Nu_L = \frac{hL}{k} \qquad (3)$$

Replacing *Nu* by the *Stanton number, St* and introducing the *Prandtl number, Pr* :

$$St = \frac{Nu_L}{Re_L Pr} \qquad (4) \qquad\qquad where: \; St = \frac{h}{\rho V c_p} \qquad (5)$$

$$Pr = \frac{c_p \mu}{k} \qquad (6)$$

The relation, assuming the pressure drop in the flow direction is zero and that $Pr = 1$ (true for most gases), now takes the form:

$$\frac{C_f}{2} = St \qquad (7)$$

This expression, known as the *Reynolds analogy*, relates key parameters of the velocity and thermal boundary layers. The accuracy of this expression depends on the noted restrictions, that the pressure gradient in the flow direction is zero and the Prandtl number equals one. It has been shown that the analogy may be applied over a wide range of Prandtl numbers if a correction is added. With this correction arises the *modified Reynolds*, or *Chilton-Colburn* analogy, shown as follows:

$$\frac{C_f}{2} = St\,Pr^{2/3} = j_H \qquad\qquad 0.6 < Pr < 60 \qquad (8)$$

where j_H is the *Colburn j factor*. For laminar flow, the modified Reynolds analogy is again only appropriate when the pressure drop in the flow direction is zero. In turbulent flow, conditions are less sensitive to the effect of pressure gradients and the equation remains valid for small pressure drops.

The benefits of this analogy lie in the ability to deduce heat transfer information from skin friction information and vice versa. A wide variety of compact heat exchanger performance data was compiled by Kays and London [Ref. 10] by way of plots of Colburn *j* factor and friction factor versus Reynolds number. The scope of this thesis includes presentation of Colburn *j* factor versus Reynolds number and does not include study of friction factor. In using this standard presentation, future correlation of friction factors in this standardized form is possible.

B. MANUFACTURER'S DATA

Plots of Colburn *j* factor versus Reynolds number, or any related information, were sought from the each of the heat exchanger manufacturers with no success. It was evident that in order to see desired heat exchanger performance characteristics, more commonly available information such as that shown in Figure 2 would have to be recast to the accepted non-dimensional j_H and *Re* forms.

Air Temperature Rise at 5 PSIG, 0° EDB
(WF or WR Alum. Fins)
Face Velocity, SFPM

Row Fin	200	400	600	800	1000	1200
104	49.7	37.6	31.8	27.8	25.3	23.2
106	68.1	51.1	42.9	37.7	34.1	31.5
108	85.2	63.6	52.6	45.8	41.2	38.0
110	101.1	74.9	61.3	53.4	47.7	43.1
112	114.9	84.7	69.7	60.0	53.1	48.0
206	115.8	90.8	77.4	69.0	62.7	58.1
208	138.5	109.0	93.1	82.9	74.9	69.0
210	156.9	124.9	106.7	94.3	85.2	78.6
212	171.5	138.1	118.1	104.2	93.9	86.3

Figure 2. Typical Manufacturer's Heat Exchanger Performance Data

C. ANALYSIS MODEL

The heat exchangers studied are represented in Figure 3:

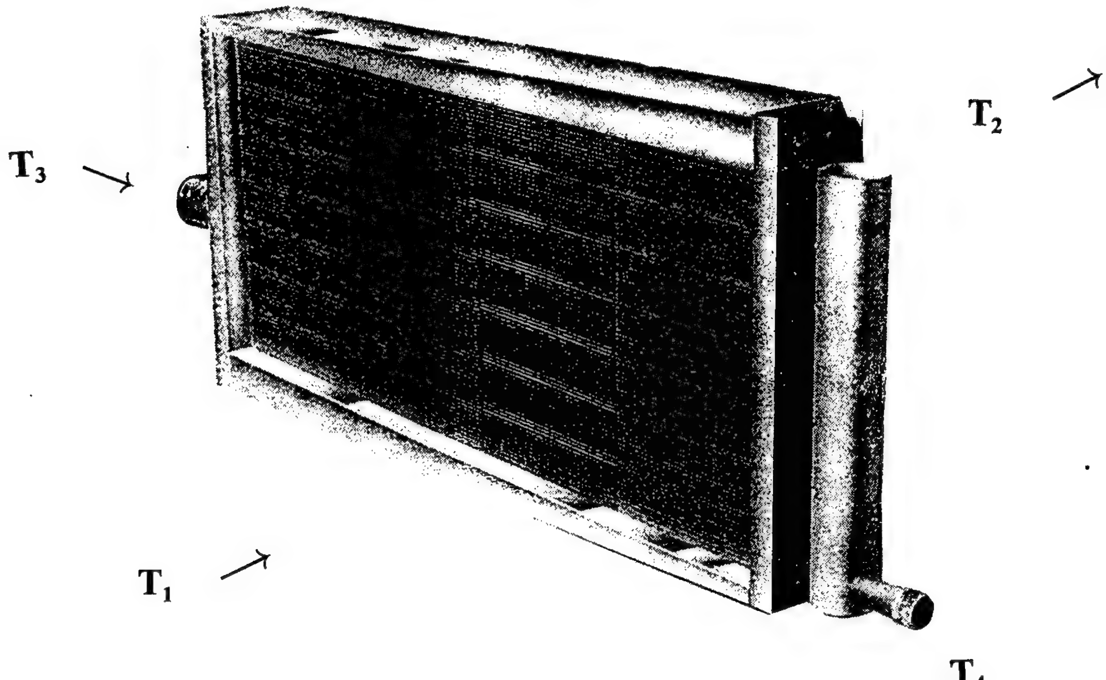


Figure 3. Heat Exchanger Representative Model

where: T_1 = Air (cold fluid) inlet temperature
T_2 = Air (cold fluid) outlet temperature
T_3 = Steam (hot fluid) inlet temperature
T_4 = Steam (hot fluid) outlet temperature

D. COMPUTATIONS AND RESULTS

1. Relation Definitions

Relations used in the steam system heat exchanger computations are defined as follows:

<u>Mean air temperature:</u>

$$T_M = \frac{(T_1 + T_2)}{2} \qquad (9)$$

Air mass flow rate:

$$\dot{m}_{air} = \rho_{air} * A_{fr} * V_{air} \tag{10}$$

Air side heat transfer rate:

$$\dot{Q}_{air} = \dot{m}_{air} * c_{p_{air}} * (T_2 - T_1) \tag{11}$$

Air side convection coefficient:

$$h_{air} = \frac{\dot{Q}_{air}}{(T_S - T_1) * A_{s_{air}}} \tag{12}$$

Mass velocity:

$$G = \frac{\dot{m}_{air}}{A_{ff}} \tag{13}$$

Reynolds number:

$$Re_{air} = \frac{G * D_h}{\mu} \tag{14}$$

Prandtl number:

$$Pr = \frac{c_p * \mu}{k} \tag{15}$$

Stanton number:

$$St = \frac{h}{G * c_p} \tag{16}$$

Colburn *j* factor:

$$j_H = St * Pr^{2/3} \tag{17}$$

2. Sequence of Computations

Computations for the steam system heat exchanger analyses were performed with assistance of Fortran computer codes. The sequence of computations in the Fortran code titled “hxair.f”, shown as part of Appendix C, is summarized in Figure 4. Manufacturer’s data for each heat exchanger similar to that shown in Figure 2 were input to the program. Dimensional characteristics of the heat exchangers input to the program were determined from manufacturer’s drawings (see Appendix B for a tabular summary of all pertinent dimensions).

3. Tabular and Graphical Results

Program output, shown in tabular form in Appendix C, included the Colburn *j* factor and Reynolds numbers needed for further analysis of a hydronic system in a subsequent chapter. Plots of j_H vs. Re for each heat exchanger studied are shown in Figures 5 through 11.

4. Comparison with Established Results

Results obtained from the steam system analysis were compared with the established results of Kays and London [Ref. 10]. For a similarly configured heat exchanger, a Colburn *j* factor versus Reynolds number plot is shown in Figure 12. It is evident that orders of magnitude of Colburn *j* factors at respective Reynolds numbers and overall trends of data for this analysis compare favorably with previously established results.

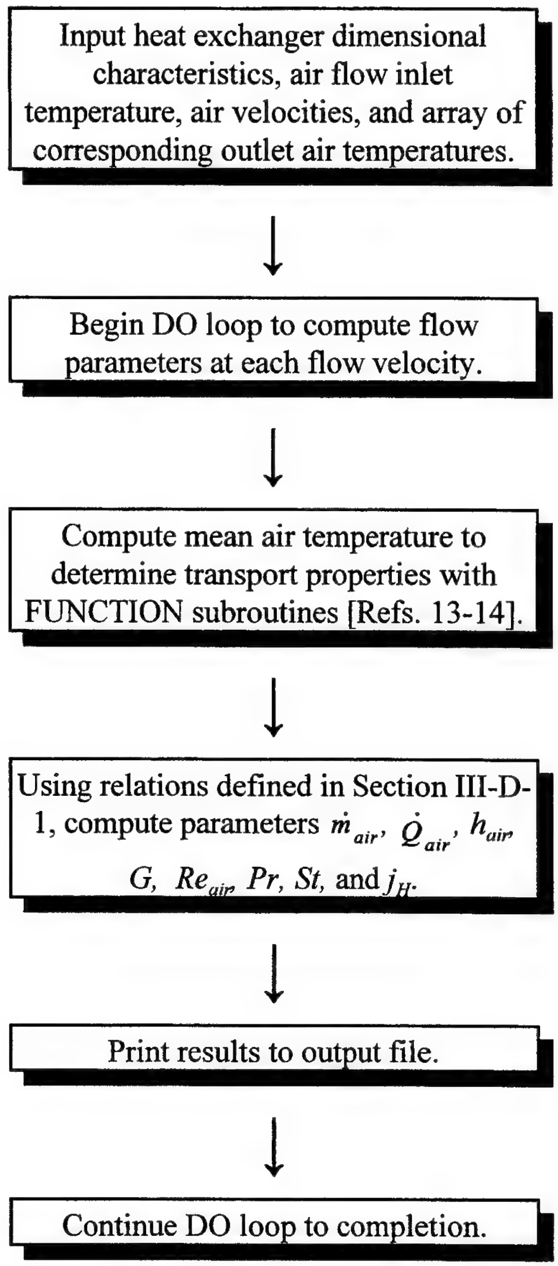


Figure 4. Steam System Computations Flow Chart

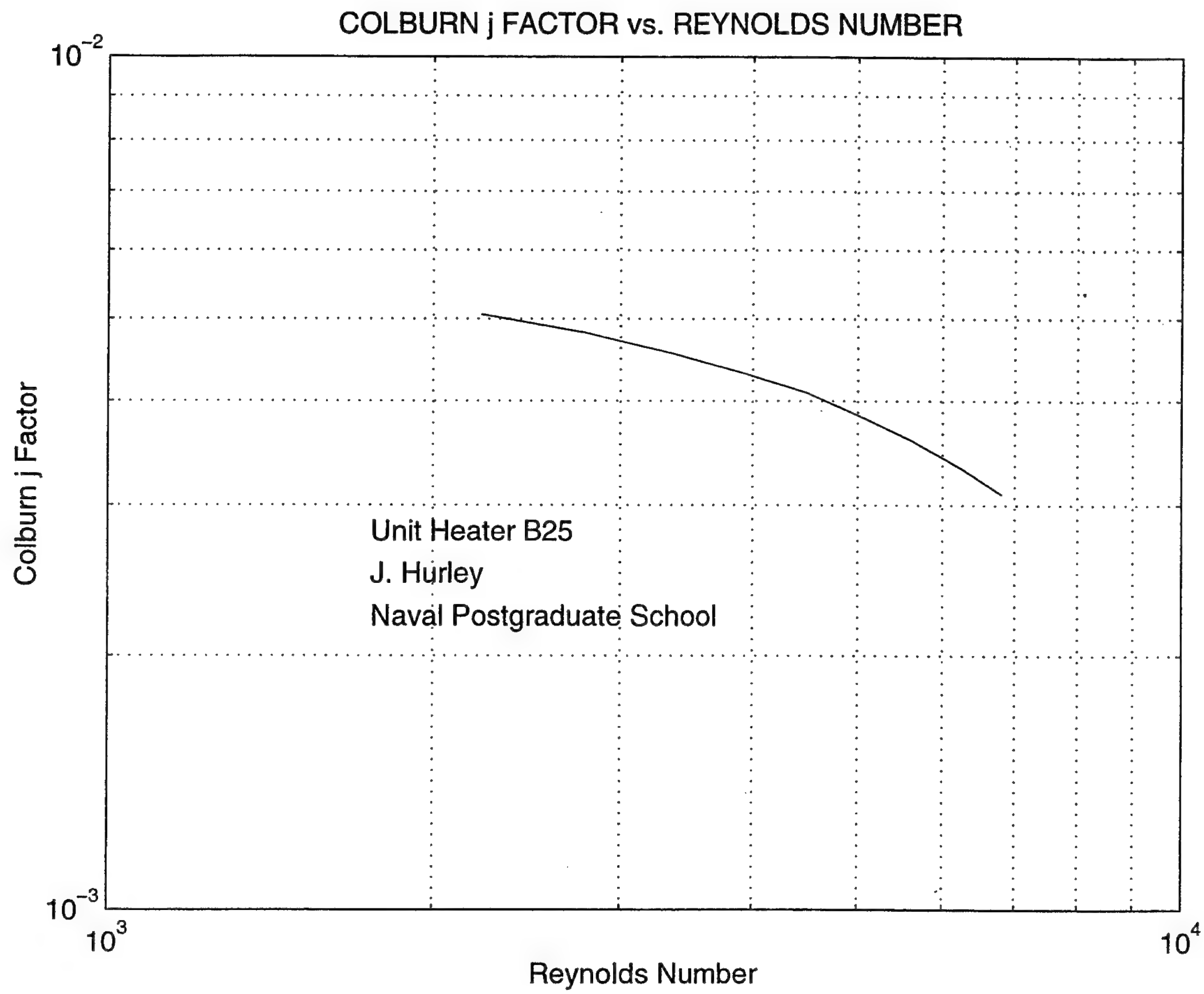


Figure 5. Colburn *j* Factor vs. Reynolds Number - Unit Heater B25

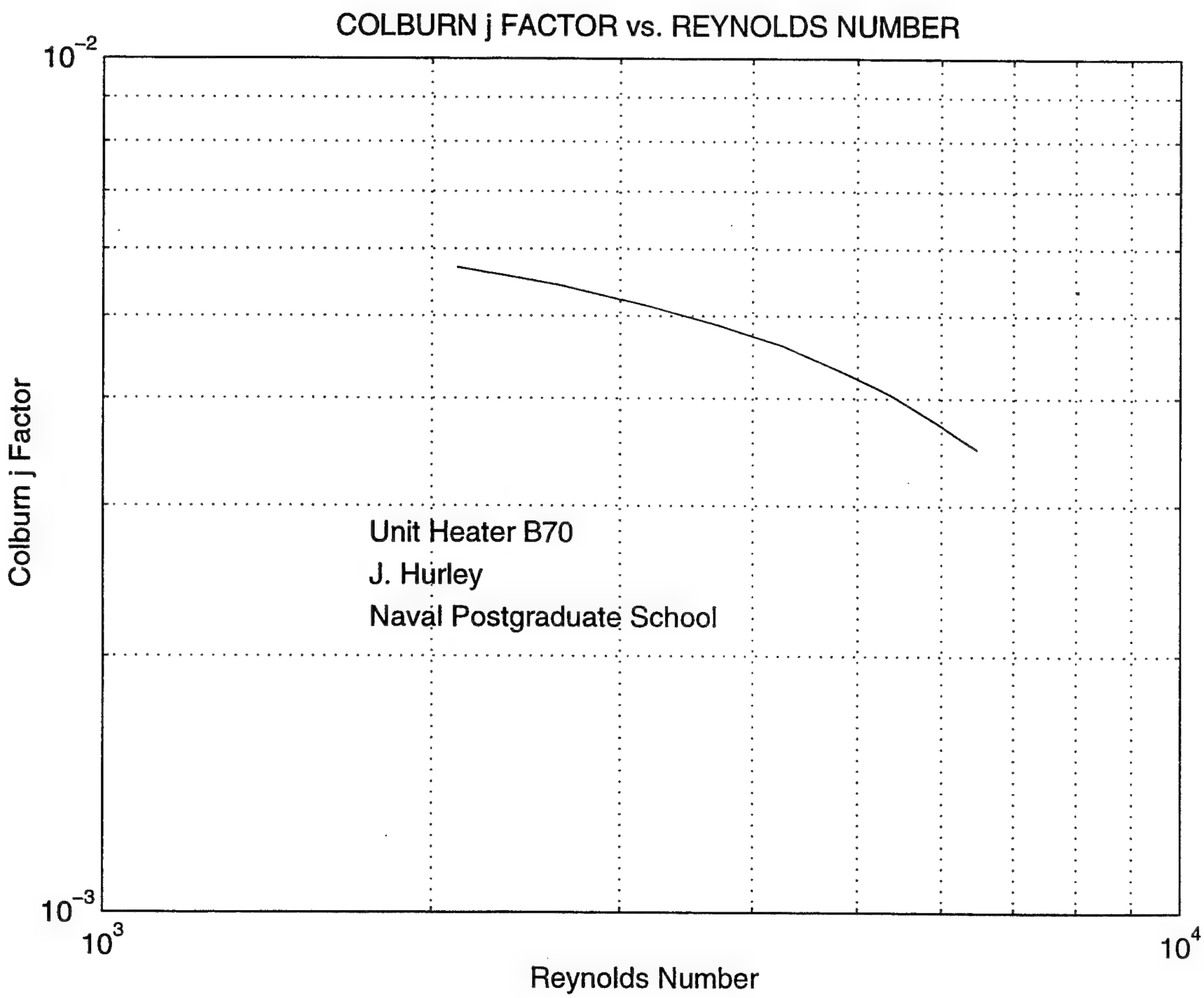


Figure 6. Colburn *j* Factor vs. Reynolds Number - Unit Heater B70

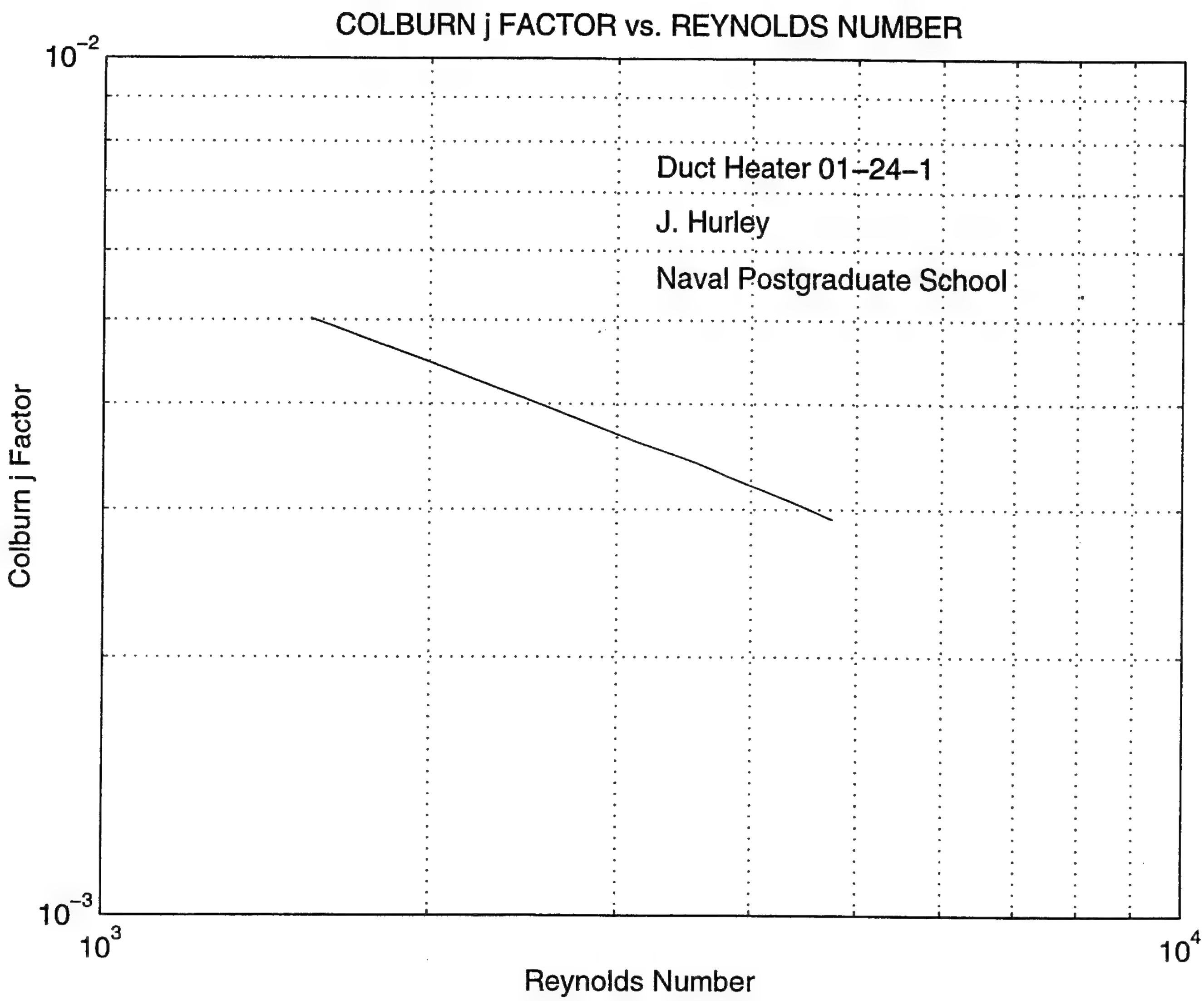


Figure 7. Colburn *j* Factor vs. Reynolds Number - Duct Heater 01-24-1

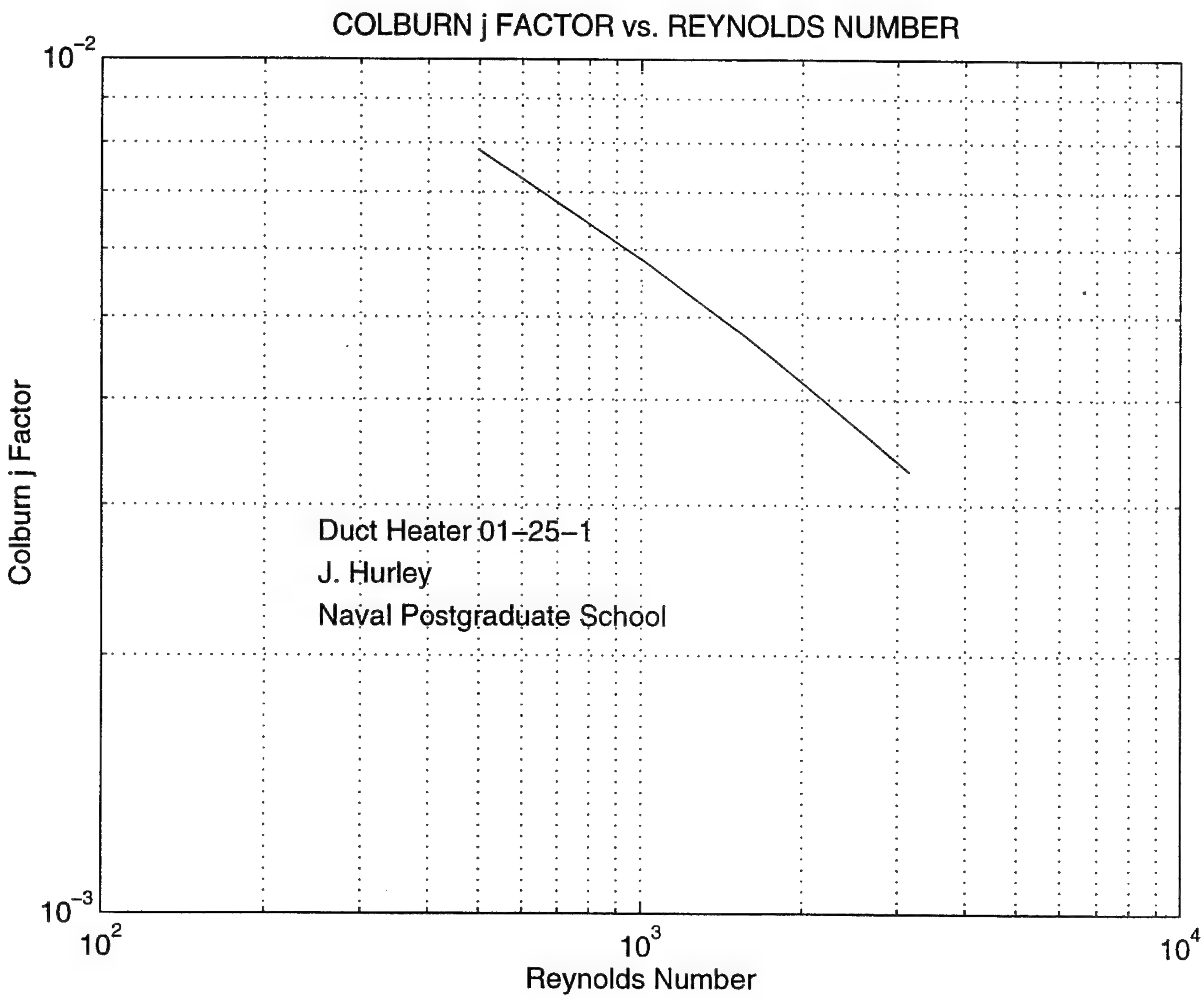


Figure 8. Colburn *j* Factor vs. Reynolds Number - Duct Heater 01-25-1

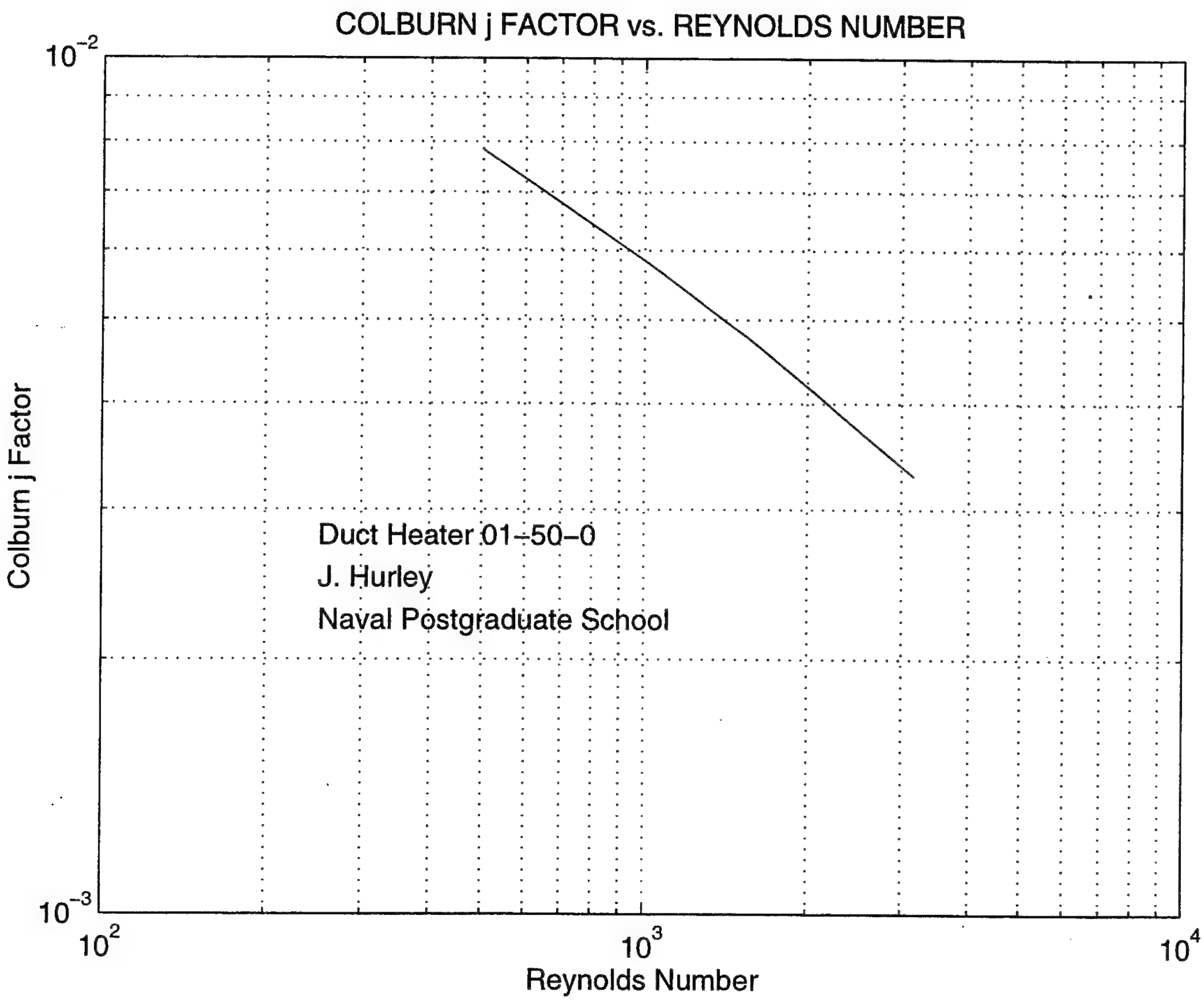


Figure 9. Colburn *j* Factor vs. Reynolds Number - Duct Heater 01-50-0

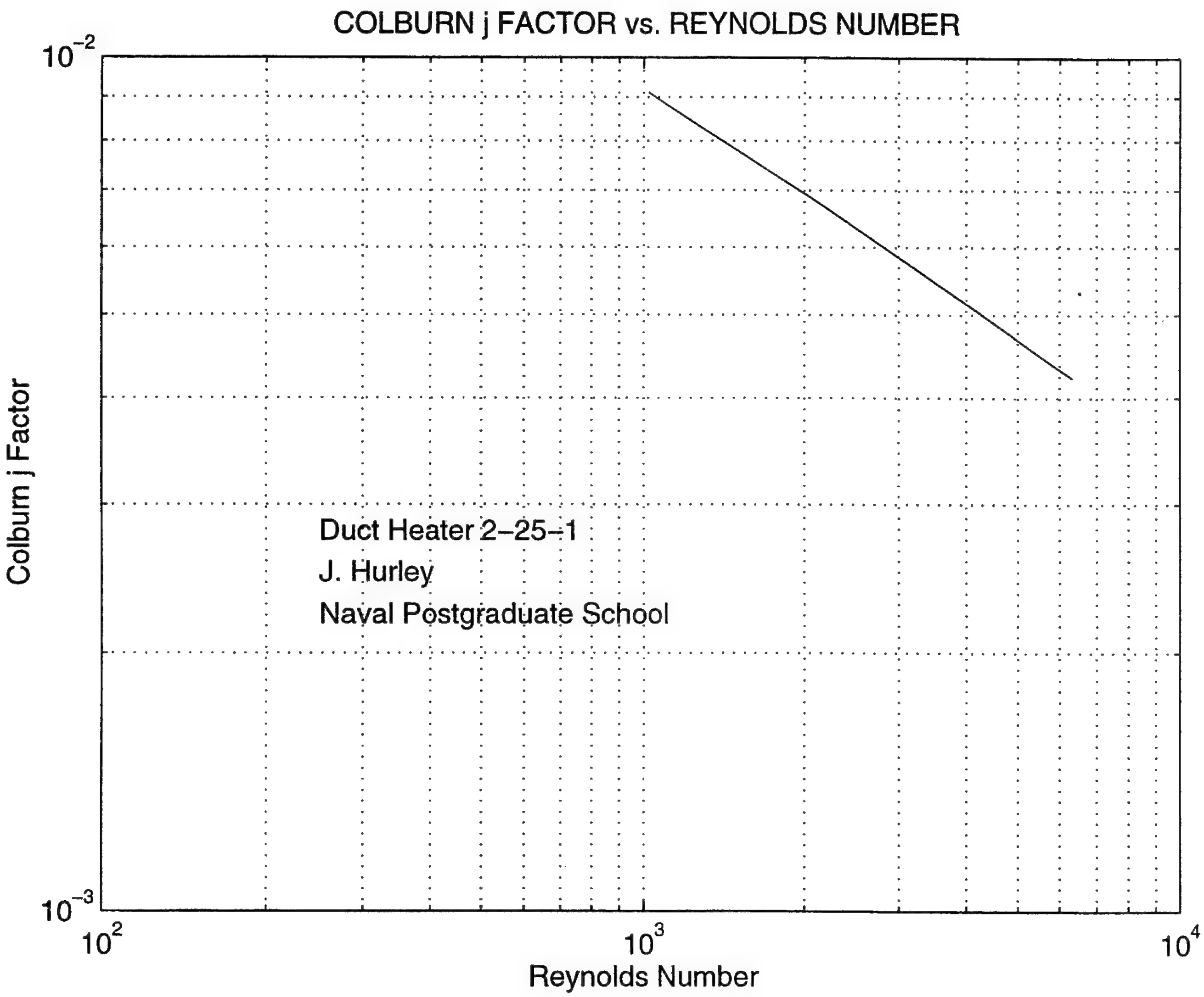


Figure 10. Colburn *j* Factor vs. Reynolds Number - Duct Heater 2-25-1

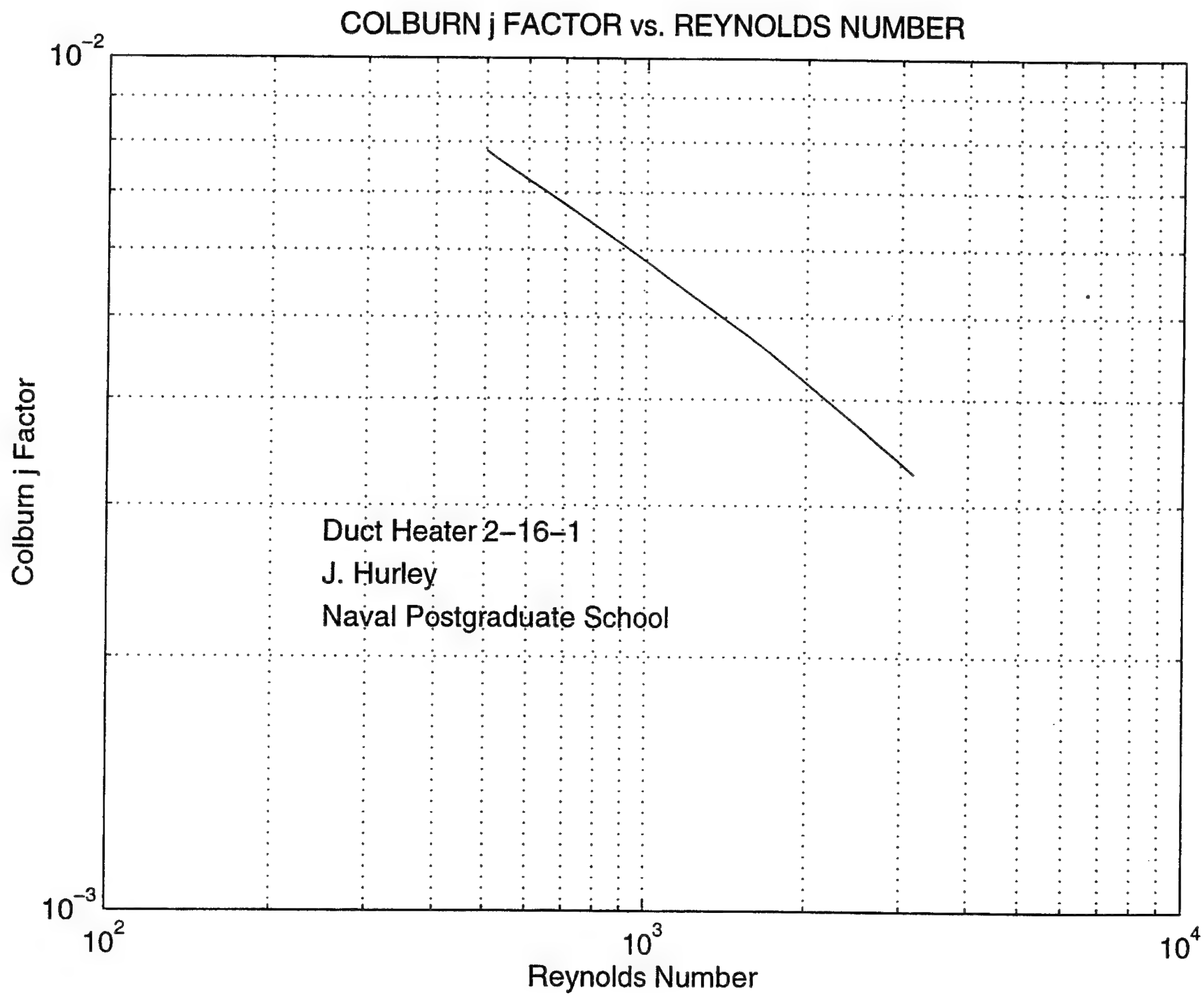
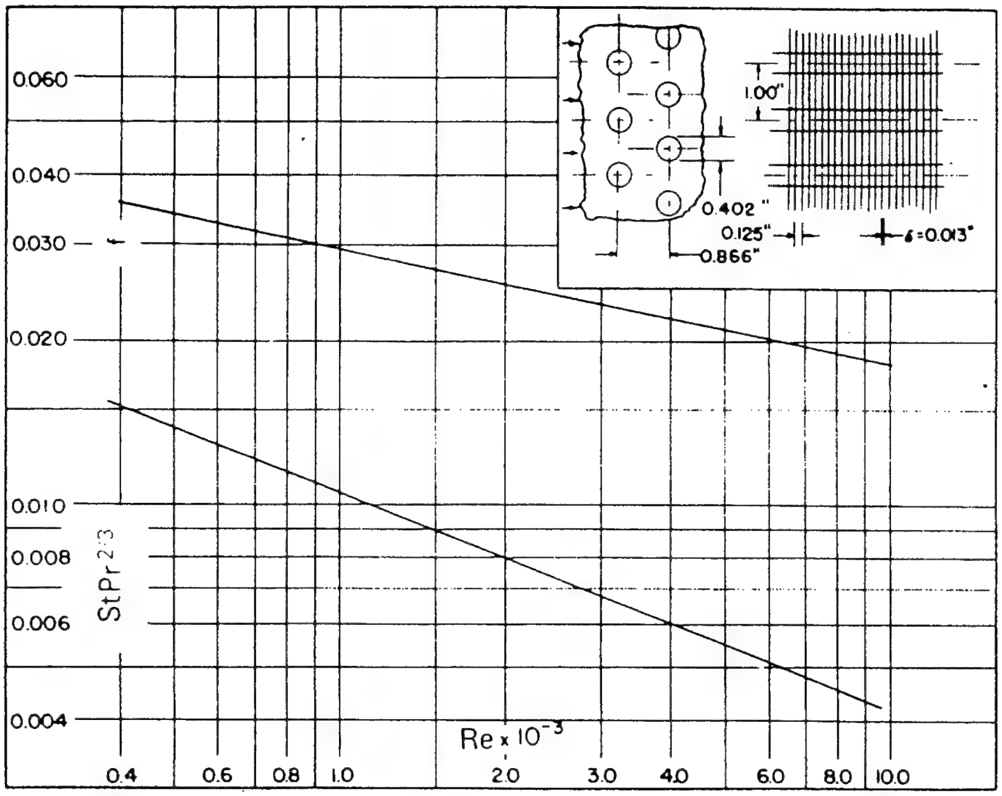


Figure 11. Colburn *j* Factor vs. Reynolds Number - Duct Heater 2-16-1

Finned circular tubes, surface 8.0–3/8T.



Tube outside diameter = 0.402 in = 10.2×10^{-3} m

Fin pitch = 8.0 per in = 315 per m

Flow passage hydraulic diameter, $4r_h$ = 0.01192 ft = 3.632×10^{-3} m

Fin thickness = 0.013 in = 0.33×10^{-3} m

Free-flow area/frontal area, σ = 0.534

Heat transfer area/total volume, α = 179 ft^2/ft^3 = 587 m^2/m^3

Fin area/total area = 0.913

Note: Minimum free-flow area in spaces transverse to flow.

Figure 12. Established Colburn *j* Factor vs. Reynolds Number Relation [After Ref. 10]

IV. HYDRONIC SYSTEM ANALYSIS

A. APPROACH

To analyze the compartment heating system as a hydronic (circulating water) system, the Effectiveness - NTU Method is utilized. Required for this method are determinations of heat exchanger overall heat transfer coefficients, which entails computing other parameters such as air and water convection coefficients, fin efficiencies, wall conduction resistances, and fouling resistances. Results from the previous chapter are an integral part of this analysis, and are used specifically to determine air side convection coefficients at specified fan operating conditions.

B. OVERALL HEAT TRANSFER COEFFICIENT

1. Newton's Law of Cooling

A standard measure of a heat exchanger's total thermal resistance to heat transfer between two fluids separated by a heat exchanger structure is the *overall heat transfer coefficient, U.* In an expression analogous to Newton's Law of Cooling, the heat transfer between two fluids separated by one or more thermal resistances, R_t, is:

$$\dot{Q} = \frac{\Delta T}{\sum R_t} = UA_s \Delta T \qquad (18)$$

The result apparent from the above expression, that $1/\sum R_t = UA_s$, is applicable to clean, unfinned surfaces only, but can be used as the basis for determining a heat exchanger's overall heat transfer coefficient with additional factors considered such as the

effects of fins on the air side surfaces, the presence of fouling on both air and water sides, and tube wall conduction.

2. Overall Coefficient Elements

a. Air Side Convection Coefficients

The *air side convection coefficients, h_{air}*, are determined using the Reynolds number vs. Colburn *j* Factor plots, generated as part of the steam system analysis in the previous chapter. The utility of presenting data in non-dimensional form is now very apparent, as the performance characteristics of each heat exchanger formulated with steam now is relevant to the hydronic analysis.

An actual operating condition of the fan supplying air to a particular heat exchanger, in the form of a quantity of cubic feet per minute of air at a specified air temperature (taken from manufacture's data) is used to compute the air flow's *mass flow rate, $\dot{m}_{air}$*, then *mass velocity, G*, then *Reynolds number, Re*. Entering the *Re* vs. j_H graph with the computed Reynolds number, a value for j_H is obtained, from which h_{air} is computed using the Stanton number and Colburn *j* factor relations, simplified into the following:

$$h_{air} = \frac{j_H \, G \, c_p}{Pr^{2/3}} \qquad (19)$$

Computations of the air side convection coefficient for each of the system's heat exchangers are shown in Table 3. Note that model B-25 unit heaters (i.e. 2-8-0 and 2-73-1) and model B-70 unit heaters (2-80-1, 2-40-1, 2-40-2, 2-60-1, 2-60-2)

each have one type fan and thus a common volumetric flow rate of air ("CFM of Air @ 70°F"), and were each grouped into one row in the table.

Heater Number	CFM of air @ 70°F	$\dot{m}$ (lbm/hr)	G (lbm/ft^2-sec)	Re	j_H	h_{air} (BTU/hr-ft^2-R)
Type B-25*	400	1,798	1.16	2,981	0.0048	6.0
Type B-70**	1,505	6,763	2.25	5,795	0.0037	9.0
01-24-1	3,680	16,538	1.63	2,762	0.0040	7.0
01-25-1	1,400	6,292	1.62	1,691	0.0047	8.2
01-50-0	750	3,371	1.16	1,211	0.0056	7.0
2-25-1	1,050	4,719	1.73	3,654	0.0054	10.1
2-16-1	350	1,573	0.82	851	0.0062	5.4

* Type B-25 includes unit heaters 2-8-0 and 2-73-1.

** Type B-70 includes unit heaters 2-80-1, 2-40-1, 2-40-2, 2-60-1, 2-60-2.

Table 3. Air Side Convection Coefficient Computations

b. Water Side Convection Coefficients

The *water side convection coefficients, h_{wtr},* are computed based on internal flow relations presented by Incropera and DeWitt [Ref. 11]. Initial computations of water side convection coefficients are based on a manufacturer's recommended maximum water flow rate of ten gallon per minute. These initial water side convection coefficients will be modified in some cases based on an optimized water mass flow rate, as will be shown in a later section. It is also noted, as it was in the previous chapter, that the water side convection coefficient is generally one or two orders of magnitude higher

than the gas side convection coefficient. The water side convection thermal resistance ($R_t = 1/hA$) will therefore be one or two orders of magnitude lower than that of the air side, and a relatively small contributor to the sum of thermal resistances used to compute the overall heat transfer coefficient, U, as will be shown in a following section.

(1) Reynolds Number for Internal Flow. The Reynolds number for internal flows can be computed using the following relation, where the *hydraulic diameter,* $D_h = 4A_c/P$.

$$Re_D = \frac{4\dot{m}_{wtr}}{\pi D_h \mu_{wtr}} \qquad (20)$$

(2) Nusselt Number. The *Nusselt Number,* Nu_D, relations for internal laminar flow ($Re_D \leq 2{,}300$) and internal turbulent flow ($Re_D > 2{,}300$ – *Gnielinski Correlation*) are then utilized to solve for h_{wtr}:

$$Nu_D = \frac{h_{wtr} D_h}{k} \qquad (21)$$

$$= 4.36 \qquad Re_D \leq 2300, \textit{ circular duct heater tubes}$$

$$= 6.49 \qquad Re_D \leq 2300, \textit{ rectangular unit heater tubes}$$

$$= \frac{(f/8)\,(Re_D - 1000)\,Pr}{1 + 12.7\,(f/8)^{1/2}\,(Pr^{2/3} - 1)} \qquad Re_D > 2300, \textit{ duct heater and unit heater tub}$$

$$\textit{where: } f = (0.79 \ln Re_D - 1.64)^{-2}$$

(3) Sequence of Computations. Computations of the water side convection coefficients for each of the system's heat exchangers were completed with assistance of Fortran computer codes. The sequence of computations in the Fortran code titled "hxwtr.f", shown as part of Appendix C, is summarized in Figure 13. Dimensional characteristics of the heat exchangers input to the program were determined from manufacturer's drawings (see Appendix B for a tabular summary of all pertinent dimensions).

(4) Tabular Results. Excerpts from program output is shown in Tables 4 through 6. Note that the water tubing dimensions for type B-25 and B-70 unit heaters are identical, and thus these results are grouped into Table 4. Similarly, tubing dimensions for all duct heaters with the exception of heater 01-24-1 are identical and these results are grouped into Table 5. Results for heater 01-24-1, with its own unique tubing dimensions are shown in Table 6. Complete program output is shown in Appendix C.

c. Extended Surfaces (Fins)

As discussed in the previous chapter, extended surfaces or fins are often added to heat exchanger surfaces to decrease thermal resistance to convection heat transfer. A representative fin array for the heat exchangers being analyzed is shown in Figure 14. Expressions for fin efficiencies are as presented by Incropera and DeWitt [Ref. 11].

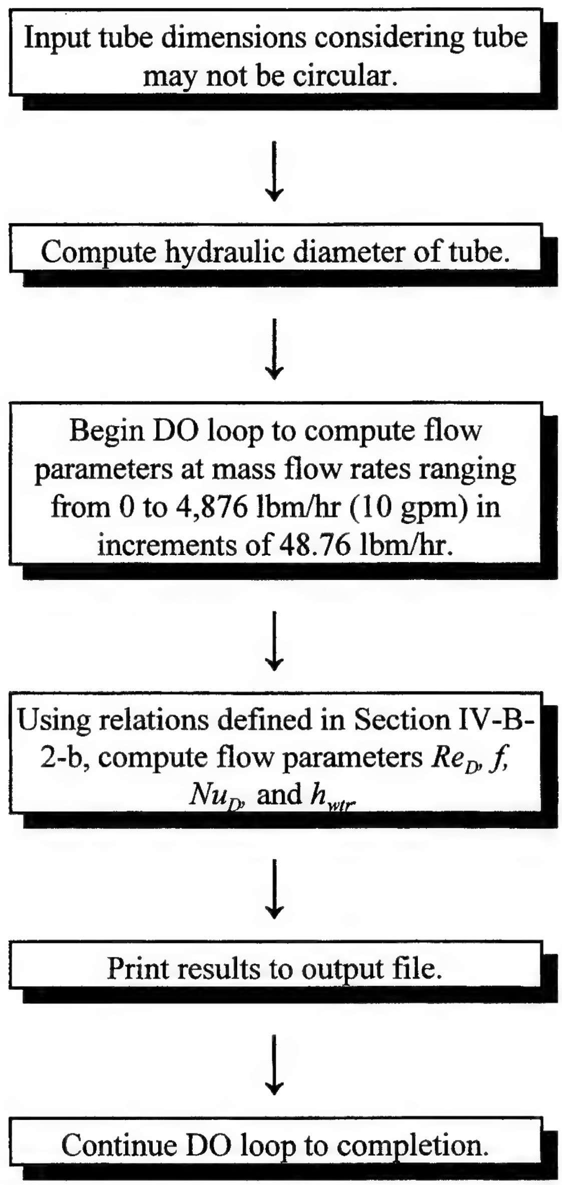


Figure 13. Water Side Convection Coefficient Computations Flow Chart

Flow Rates Mass (lbm/hr)	Volume (gal/min)	Re_D	Nu_D	h_{wtr} (BTU/hr-ft^2-R)
48.8	0.100	1482.9	6.49	53.4
97.5	0.200	2965.8	15.13	124.4
146.3	0.300	4448.7	23.88	196.4
780.2	1.600	23726.5	107.45	883.6
828.9	1.700	25209.4	113.04	929.6
4876.0	10.00	148290.7	492.92	4053.2

Table 4. Unit Heater Water Side Convection Coefficient Computations

Flow Rates Mass (lbm/hr)	Volume (gal/min)	Re_D	Nu_D	h_{wtr} (BTU/hr-ft^2-R)
48.8	0.100	1505.1	4.36	36.4
97.5	0.200	3010.2	15.41	128.6
146.3	0.300	4515.3	24.25	202.4
780.2	1.600	24081.5	108.80	908.0
828.9	1.700	25586.6	114.46	955.3
4876.0	10.00	150509.5	499.06	4165.2

Table 5. Duct Heater Water Side Convection Coefficient Computations (with exception of heater 01-24-1)

Flow Rates Mass (lbm/hr)	Flow Rates Volume (gal/min)	Re_D	Nu_D	h_{wtr} (BTU/hr-ft^2-R)
48.8	0.100	807.1	4.36	19.5
97.5	0.200	1614.2	4.36	19.5
146.3	0.300	2421.2	11.55	51.7
780.2	1.600	12913.3	64.15	287.1
828.9	1.700	13720.4	67.58	302.5
4876.0	10.00	80708.0	297.17	1329.9

Table 6. Duct Heater 01-24-1 Water Side Convection Coefficient Computations

(1) Single Fin Efficiency. The *single fin efficiency,* η_f, is given by the expression:

$$\eta_f = \frac{\tanh mL_{fin_c}}{mL_{fin_c}} \tag{22}$$

where:

$$m = \sqrt{\frac{h_{air} P}{k_{fin} A_{c_{fin}}}} \tag{23}$$

$$L_{fin_c} = L_{fin} + \frac{t_{fin}}{2} \tag{24}$$

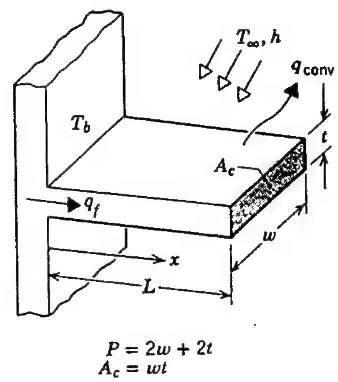


Figure 14. Representative Fin Array

(2) Fin Overall Surface Efficiency. Characterizing the performance of an array of fins is done with the expression *fin overall surface efficiency,* η_o, where:

$$\eta_o = \frac{q_t}{q_{\max}} = \frac{q_t}{hA_{s_{air}}(T_{base} - T_\infty)} \qquad (25)$$

Breaking down the total heat transfer into contributions from the finned and unfinned (base) surfaces and noting that $A_{s_{air}} = A_{s_{fin}} + A_{base}$, this expression can be further simplified yielding:

$$\eta_o = 1 - \frac{A_{s_{fin}}}{A_{s_{air}}}(1 - \eta_f) \qquad (26)$$

(3) Sequence of Computations. Computations of single and overall fin efficiencies for each of the system's heat exchangers were completed with assistance of Fortran computer codes. The sequence of computations in the Fortran code

titled "hxU.f", shown as part of Appendix C, is summarized in Figure 15. Dimensional characteristics of the heat exchangers input to the program were determined from manufacturer's drawings (see Appendix B for a tabular summary of all pertinent dimensions).

(4) Tabular Results. A summary of program output for each of the system's heat exchangers is shown in Table 7. Efficiencies were computed using operating conditions and air side convection coefficients of Table 3. Complete program output is shown in Appendix C.

Heater	η_f	η_o
B-25* Unit Heaters	0.87	0.89
B-70** Unit Heaters	0.82	0.84
01-24-1 Duct Heater	0.92	0.93
01-25-1 Duct Heater	0.90	0.90
01-50-0 Duct Heater	0.91	0.92
2-25-1 Duct Heater	0.92	0.92
2-16-1 Duct Heater	0.93	0.93

Table 7. Heater Single Fin and Overall Fin Efficiencies

d. Fouling Resistance

During normal heat exchanger operation, surfaces are often subject to fouling by fluid impurities, rust or scale formation, and other reactions between the fluid

and the wall material. Depositions such as these can greatly increase the resistance to heat transfer between the fluids. This effect can be treated by introducing an additional thermal resistance, termed the *fouling factor*, R_f, the value of which depends on the operating temperature, fluid velocity, and length of service of the heat exchanger. Fouling factors recommended by the Tubular Exchanger Manufacturer's Association are shown below in Table 8. The shaded fouling factors are those used for the heating system analyzed in this thesis.

e. Tube Wall Conduction

Derivations of the *tube wall conduction resistance,* R_{wall} by Incropera and

Fluid	Fouling Factor (hr-ft^2-°F/BTU)	
	Below 125 °F	Above 125 °F
Seawater	0.0005	0.001
Distilled Water	0.0005	0.0005
Treated Boiler Feedwater	0.001	0.001
City or Well Water	0.001	0.002
Hard Water	0.003	0.005
Air	0.002	0.002
Diesel Exhaust	0.01	0.01
Clean Steam	0.0005	0.0005

Table 8. Typical Fouling Factors

DeWitt [Ref. 11] produce the following expressions for duct heater tubes (cylindrical) and unit heater tubes (modeled as plane wall).

$$R_{wall} = \frac{\ln\left(\frac{d_o}{d_i}\right)}{(2\pi L k)_{tube}} \tag{27}$$

(cylindrical tube wall conduction resistance)

$$R_{wall} = \left(\frac{t}{kA_s}\right)_{tube} \tag{28}$$

(plane wall tube conduction resistance)

3. Summation of Elements

The summation of thermal resistances including effects of air side convection with fins, water side convection without fins, fouling on both air and water sides, and tube wall conduction yields the following:

$$\begin{aligned} \frac{1}{UA_s} &= \frac{1}{U_{air}A_{s_{air}}} = \frac{1}{U_{wtr}A_{s_{wtr}}} \\ &= \frac{1}{(\eta_o h A_s)_{air}} + \frac{R_{f_{air}}}{(\eta_o A_s)_{air}} + R_{wall} + \frac{R_{f_{wtr}}}{A_{s_{wtr}}} + \frac{1}{(hA_s)_{wtr}} \end{aligned} \tag{29}$$

The calculation of the UA_s product does not require designation of the air side or water side, i.e. $UA_s = U_{air}A_{s_{air}} = U_{wtr}A_{s_{wtr}}$. However, the calculation of an overall coefficient, U, does depend on whether it is based on the air side or the water side surface area since $U_{air} \neq U_{wtr} \;\; if \;\; A_{s_{air}} \neq A_{s_{wtr}}$. As mentioned previously, the convection

coefficient for the air side is generally much smaller than that for the water side, and thus dominates determination of the overall coefficient.

4. Overall Coefficient Computations and Results

a. Sequence of Computations

Computations of the overall coefficients for each of the system's heat exchangers were completed with assistance of Fortran computer codes. The sequence of computations in the Fortran code titled "hxU.f", shown as part of Appendix C, is summarized in Figure 15. Program inputs are the previously computed values of h_{air}, h_{wtr}, tabulated values of $R_{f_{air}}$ and $R_{f_{wtr}}$, and dimensional and material characteristics (see Appendix B for a tabular summary of all pertinent dimensions).

b. Tabular Results

A summary of program output for each of the system's heat exchangers is shown in Table 9. Complete program output is shown in Appendix C.

C. THE EFFECTIVENESS - NTU METHOD

1. Applicability

The *Effectiveness - NTU* analysis method is used to take the overall heat transfer coefficient parameter and determine heat exchanger heat transfer performance characteristics. This method is chosen over the *log-mean temperature difference* method based on a detailed comparison between the two methods presented by Kays and London [Ref. 10]. In the comparison, steps for each method are evaluated for a variety of

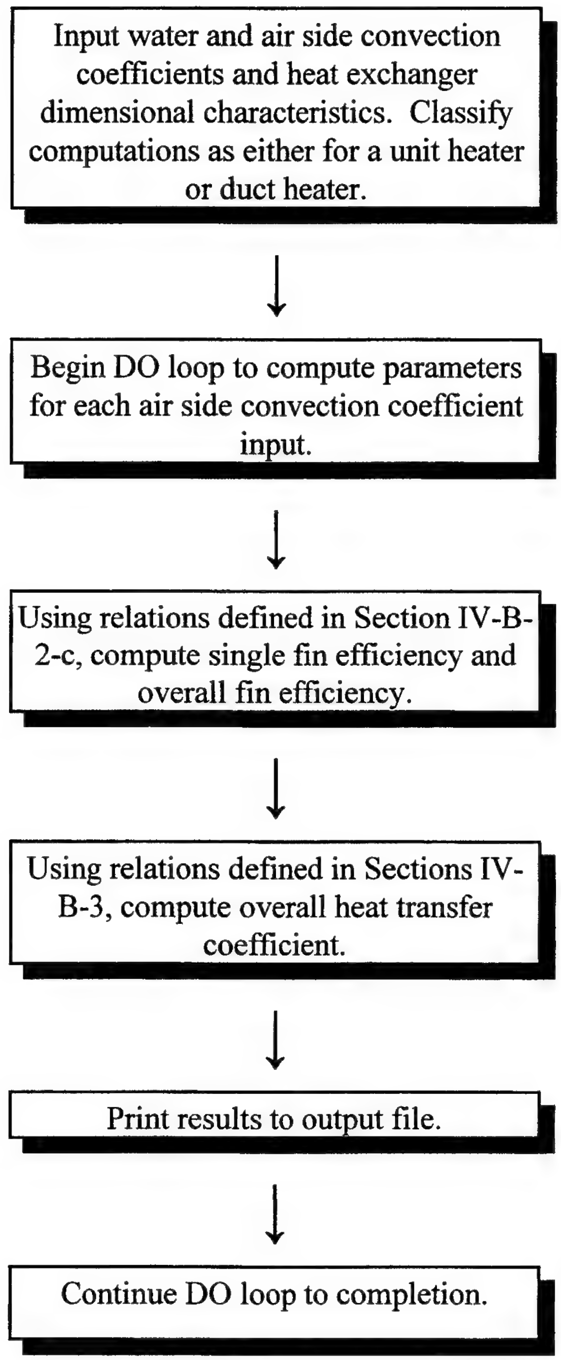


Figure 15. Fin Efficiency and Overall Coefficient Computations Flow Chart

Heater	U_{air} (BTU/hr-ft^2-R)
B-25 Unit Heaters*	4.9
B-70 Unit Heaters**	6.7
01-24-1 Duct Heater	5.7
01-25-1 Duct Heater	6.4
01-50-0 Duct Heater	5.6
2-25-1 Duct Heater	8.3
2-16-1 Duct Heater	4.5

* Type B-25 include unit heaters 2-8-0 and 2-73-1.

** Type B-70 include unit heaters 2-80-1, 2-40-1, 2-40-2, 2-60-1, 2-60-2.

Table 9. Heat Exchanger Overall Coefficient Results

analysis starting points. For this analysis, where only the inlet hot and cold fluid temperatures are known, the Effectiveness - NTU method is more straightforward, eliminating a tedious, iterative procedure where outlet temperatures would be estimated and then adjusted until the heat transfer rate corresponds to the inlet/outlet temperature difference. In the Effectiveness - NTU (abbreviation for "number of transfer units") approach, from knowledge of the heat exchanger type, size, and fluid flow rate, the NTU, *maximum heat transfer rate,* $\dot{Q}_{\max}$, and *effectiveness,* ϵ (both defined shortly) are used to determine the actual heat transfer rate.

2. Maximum Possible Heat Transfer Rate

The maximum possible heat transfer rate for a given heat exchanger is achieved when, by definition, the entering cold air temperature is raised to equal its highest possible value, i.e., that of the entering hot water temperature. Referring to Figure 3, entering air would rise in temperature by the quantity $(T_3 - T_1)$. Using a relationship for the heat transfer rate similar to that presented earlier:

$$\dot{Q} = \dot{m}c_p(T_3 - T_1) \qquad (30)$$

where the term $\dot{m}c_p$ is defined as the *heat capacity rate, C*, and will be either based on the mass flow rate, $\dot{m}$, and specific heat, c_p, for air or for water. As developed by Incropera and DeWitt [Ref. 11], the *maximum heat transfer rate,* $\dot{Q}_{\max}$, is then:

$$\dot{Q}_{\max} = C_{\min}(T_3 - T_1) \qquad (31)$$

where $C_{\min}$ is equal to either C_{air} or C_{wtr}, whichever is smaller.

3. Effectiveness

The *effectiveness,* ϵ, is defined as the ratio of the actual heat transfer rate for a heat exchanger to the maximum possible heat transfer rate.

$$\epsilon = \frac{\dot{Q}}{\dot{Q}_{\max}} \qquad (32)$$

The effectiveness, which is dimensionless, must be in the range $0 \le \epsilon \le 1$. It is

useful because, if ϵ, T_3, and T_1 are known, the actual heat transfer rate may be determined from the expression:

$$\dot{Q} = \epsilon \dot{Q}_{max} = \epsilon C_{min}(T_3 - T_1) \qquad (33)$$

4. Number of Transfer Units

The *number of transfer units*, *NTU*, is a dimensionless parameter that is widely used for heat exchanger analysis and is defined as:

$$NTU = \frac{UA_s}{C_{min}} \qquad (34)$$

5. Effectiveness - NTU Relations

The Effectiveness - NTU relationship for the type heat exchangers being analyzed, i.e. a single pass cross flow heat exchanger with both fluids unmixed is:

$$\epsilon = 1 - \exp\left[\left(\frac{1}{C_r}\right)(NTU)^{0.22}\left\{\exp\left[-C_r(NTU)^{0.78}\right] - 1\right\}\right] \qquad (35)$$

where:

$$C_r = \frac{C_{min}}{C_{max}} \qquad (36)$$

The *NTU*, ϵ, C_r relation may be represented graphically as shown in Figure 16.

6. Computations and Results

a. Sequence of Computations

Computations involving the Effectiveness - NTU analysis of the system's

heat exchangers were completed utilizing a spreadsheet format, shown in Appendix D. Inputs are heat exchanger dimensions, water and air inlet temperatures T_1 and T_3 respectively, air and water volumetric flow rates, and previously determined values of j_H,

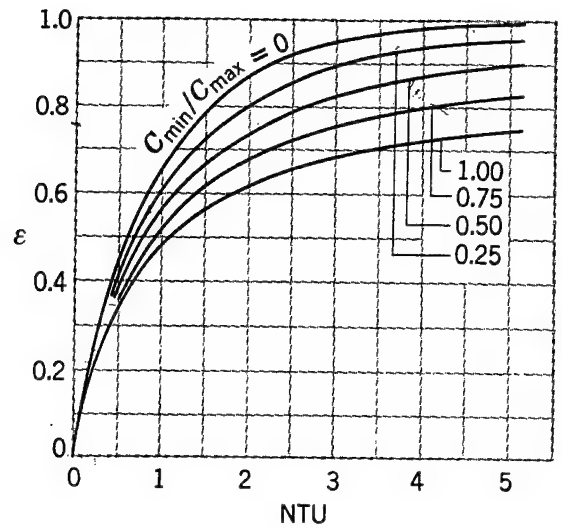


Figure 16. *NTU,* ϵ, C_r Relation of a Single-Pass Cross Flow Heat Exchanger with Both Fluids Unmixed

h_{air}, h_{wtr} *and* U_{air}. Using relations defined in Sections IV-C-2 through IV-C-5, NTU-effectiveness parameters were computed and finally the heat transfer rate and outlet air and water temperatures T_2 and T_4 were determined.

b. Tabular Results

Effectiveness - NTU analysis results for the hydronic system are shown in Table 10.

Heater	Air Flow Parameters							Water Flow Parameters										
	CFM	$\dot{m}$ $\left(\frac{lbm}{hr}\right)$	G $\left(\frac{lbm}{ft^2\,sec}\right)$	Re	J_H	h $\left(\frac{BTU}{hr\,ft^2\,R}\right)$	C $\left(\frac{BTU}{hr\,R}\right)$	$\dot{m}$ $\left(\frac{lbm}{hr}\right)$	h $\left(\frac{BTU}{hr\,ft^2\,R}\right)$	C $\left(\frac{BTU}{hr\,R}\right)$	U_{air} $\left(\frac{BTU}{hr\,ft^2\,R}\right)$	$\dot{Q}_{max}$ $\left(\frac{BTU}{hr}\right)$	NTU	C_r	ϵ	$\dot{Q}$ $\left(\frac{BTU}{hr}\right)$	T_2 (°F)	T_4 (°F)
B-25	400	1798	1.16	2981	0.0048	6.0	431	4876	4053	4886	4.9	56085	0.24	0.09	0.21	11713	87	188
B-70	1505	6763	2.25	5795	0.0037	9.0	1623	4876	4053	4886	6.7	211020	0.15	0.33	0.13	28214	77	184
01-24-1	3680	16538	1.63	2762	0.0040	7.0	3969	4876	4053	4886	5.7	508045	0.18	0.81	0.15	74778	81	175
01-25-1	1400	6292	1.62	1691	0.0047	8.2	1510	4876	4053	4886	6.4	332196	0.23	0.31	0.20	66393	14	176
01-50-0	750	3371	1.16	1211	0.0056	7.0	809	4876	4053	4886	5.6	177962	0.29	0.17	0.24	43215	23	181
2-25-1	1050	4719	1.73	3654	0.0054	10.1	1132	4876	4053	4886	8.3	169873	0.15	0.23	0.13	22526	60	185
2-16-1	350	1573	0.82	851	0.0062	5.4	377	4876	4053	4886	4.5	54737	0.33	0.08	0.28	15152	85	187

Table 10: Effectiveness - NTU Analysis Results - Hydronic System

V. SYSTEM COMPARISONS AND DISCUSSION

A. OUTLINE

The hydronic system heat exchanger performance data generated in the previous chapter will be compared with steam system performance parameters. Specifically, the heat rate outputs ($\dot{Q}$) and the final air temperatures (T_2) of the hydronic system will be compared with those of the steam system. In cases where the hydronic system performance matches or exceeds steam system performance parameters stated on the ship's prints, water flow rates will be optimized. In instances where the hydronic system performance falls short of that of the steam system, further analysis will be conducted to attempt to increase hydronic system heat transfer.

B. FURTHER EXAMINATION OF STEAM SYSTEM CAPABILITIES

1. Modifications to Effectiveness-NTU Method

Steam system heat exchanger heat transfer performance data is essential in order to assess the capabilities of the hydronic system. To compute steam system performance parameters, the Effectiveness-NTU analysis method is again used, modified for steam as follows:

1. The convection coefficient for steam is taken to be very large (on the order of 10^4), thereby decreasing the air side thermal resistance, contributing negligibly to the overall heat transfer coefficient.

2. The specific heat for a substance undergoing a phase change, such as steam condensing, is infinite. The heat capacity rate for steam, C_{steam}, is therefore also infinite, yielding a heat capacity ratio, C_r, effectively equal to zero.
3. In the case of $C_r = 0$, the effectiveness, $\epsilon = 1 - \exp(-NTU)$.
4. The inlet air and steam temperatures are as stated previously in Chapter II.
5. The steam mass flow rate, $\dot{m}_{steam} = \dot{Q}/(h_{fg})$, where h_{fg} is the enthalpy difference due to condensation from saturated steam to saturated water.

2. Sequence of Computations

Computations involving the Effectiveness-NTU analysis of the system's heat exchangers were completed utilizing a spreadsheet format shown in Appendix D. Inputs are heat exchanger dimensions, steam and air inlet temperatures, T_1 and T_3 respectively, as well as air volumetric flow rates, and previously determined values of j_H, h_{air}, and U_{air}. Using relations defined in Chapter III, Effectiveness-NTU parameters were computed and finally the heat transfer rate and outlet air temperature T_2 for each heat exchanger were determined.

3. Tabular Results

Effectiveness-NTU analysis results for the steam system are shown in Table 11.

Heater	Air Flow Parameters							Steam Flow Parameters			U_{air}	$\dot{Q}_{max}$	NTU	C_r	ϵ	$\dot{Q}$	T_2	T_4
	CFM	$\dot{m}$ $\left(\frac{lbm}{hr}\right)$	G $\left(\frac{lbm}{ft^2\,sec}\right)$	Re	J_H	h $\left(\frac{BTU}{hr\,ft^2\,R}\right)$	C $\left(\frac{BTU}{hr\,R}\right)$	$\dot{m}$ $\left(\frac{lbm}{hr}\right)$	h $\left(\frac{BTU}{hr\,ft^2\,R}\right)$	C $\left(\frac{BTU}{hr\,R}\right)$	$\left(\frac{BTU}{hr\,ft^2\,R}\right)$	$\left(\frac{BTU}{hr}\right)$				$\left(\frac{BTU}{hr}\right)$	(°F)	(°F)
B-25	400	1798	1.16	2981	0.0048	6.0	431	17.6	>>0	∞	5.0	77656	0.24	0.00	0.22	16722	99	240
B-70	1505	6763	2.25	5795	0.0037	9.0	1623	42.5	>>0	∞	6.7	292182	0.15	0.00	0.14	40445	85	240
01-24-1	3680	16538	1.63	2762	0.0040	7.0	3969	127.7	>>0	∞	6.1	706500	0.19	0.00	0.17	121615	93	240
01-25-1	1400	6292	1.62	1691	0.0047	8.2	1510	91.9	>>0	∞	6.6	407696	0.24	0.00	0.21	87498	28	240
01-50-0	750	3371	1.16	1211	0.0056	7.0	809	59.0	>>0	∞	5.8	218408	0.30	0.00	0.26	56155	39	240
2-25-1	1050	4719	1.73	3654	0.0054	10.1	1132	33.0	>>0	∞	8.5	226498	0.15	0.00	0.14	31449	68	240
2-16-1	350	1573	0.82	851	0.0062	5.4	377	22.5	>>0	∞	4.7	73612	0.34	0.00	0.29	21428	102	240

Table 11: Effectiveness-NTU Analysis Results - Steam System

C. COMPARISONS OF HYDRONIC AND STEAM SYSTEMS

1. Tabular Results

Comparisons of the hydronic and steam systems' heat transfer performances are shown in Tables 12 and 13.

Heater	$\dot{Q}_{hydronic}$ (BTU/hr)	$\dot{Q}_{steam}$ (BTU/hr)	$\frac{\dot{Q}_{hydronic}}{\dot{Q}_{steam}}$
B-25	11,713	16,722	0.70
B-70	28,214	40,445	0.70

Table 12: Unit Heater Hydronic vs. Steam Performance Data

Heater	$\dot{Q}_{hydronic}$ (BTU/hr)	$\Delta T_{air_{hydronic}}$ (°F)	$\dot{Q}_{steam}$ (BTU/hr)	$\Delta T_{air_{steam}}$ (°F)	$\frac{\dot{Q}_{hydronic}}{\dot{Q}_{steam}}$	$\frac{\Delta T_{air_{hydronic}}}{\Delta T_{air_{steam}}}$
01-24-1	74,778	19	121,615	31	0.61	0.61
01-25-1	66,393	44	87,498	58	0.76	0.76
01-50-0	43,215	53	56,155	69	0.77	0.77
2-25-1	22,526	20	31,449	28	0.71	0.71
2-16-1	15,152	40	21,428	57	0.70	0.70

Table 13: Duct Heater Hydronic vs. Steam Performance Data

2. Comparison Trends

Comparisons between hydronic and steam system results bear some consistencies. Notably, the ratio of the hydronic to steam heat performance values average approximately 70 percent. This is a performance difference which warrants examination to determine what the possibilities are for improved performances and what are the causes for the shortfalls.

D. IMPROVING PERFORMANCE OF HYDRONIC SYSTEM

1. Altering Air Mass Flow Rate

a. Analysis Method

The Effectiveness-NTU analysis makes very apparent that the air-water heat exchangers studied are "air side limited". This can be explained qualitatively by examining, for given heat exchanger dimensions and flow conditions, the relative magnitudes of the contributors to the basic relations used in the analysis. The inverse relationships of thermal resistances in the computation of overall coefficient, U, heavily skews the resulting value of U to the lower convection coefficient in the equation , in this case h_{air}. The number of transfer units, NTU, is directly proportional to U, thus a lower value of U results in a lower NTU. Examining the graphical representation of the NTU-ϵ-C_r relation (see Figure 16), lower values of NTU yield lower values of effectiveness ϵ. The heat transfer rate, $\dot{Q}$, is then the product of $\dot{Q}_{\max}$ and ϵ.

A means to increase the air side convection coefficient, h_{air}, and thereby increasing the overall coefficient and the heat transfer rate, is to increase the rate of air

flow passing over the heat exchanger. A change in the air side convection coefficient with increasing air flow rate can be predicted by examining the plots of Colburn *j* factor vs. Reynolds number and the relation from which h_{air} is derived, i.e. $h_{air} = (j_H G c_p)/(Pr^{2/3})$. The negative slope of the plots indicate that j_H decreases with increasing Reynolds number, but it is also true that the mass velocity, *G*, increases with increasing Reynolds number. The greatest increases in h_{air} will be realized when *Re* and therefore *G* increase with minimal decrease in j_H. This leads to the qualitative deduction that increases in h_{air} are more profound if the plot of Colburn *j* factor vs. Reynolds number is nearer to horizontal.

b. Analysis Results

The net effects on h_{air} of the counteracting trends of j_H and *G*, and the effects of the plot's slope can best be seen with sample computations. Results of Effectiveness-NTU analysis computations at air flow rates of up to twice the original flow rates for duct heater B-25 are shown in Table 14. Computations for the remaining heat exchangers are shown in tables in Appendix D. Plots of heat transfer rate vs. air flow rate for each heat exchanger are shown in Figures 17 through 23.

c. Air Temperature Rise Considerations

The increased heat transfer rate at increased air flow rate produces a decrease in the rise in air temperature. This can be explained by examining the relationship between the heat transfer rate and the change in air temperature presented in a previous chapter, i.e $T_2 = T_1 + (\dot{Q}/C_{air})$. The heat transfer rate , $\dot{Q}$, increases but the

CFM	Air Flow Parameters $\dot{m}$ $\left(\frac{lbm}{hr}\right)$	G $\left(\frac{lbm}{ft^2 sec}\right)$	Re	J_H	h $\left(\frac{BTU}{hr\ ft^2\ R}\right)$	C $\left(\frac{BTU}{hr\ R}\right)$	Water Flow Parameters $\dot{m}$ $\left(\frac{lbm}{hr}\right)$	h $\left(\frac{BTU}{hr\ ft^2\ R}\right)$	C $\left(\frac{BTU}{hr\ R}\right)$	U_{air} $\left(\frac{BTU}{hr\ ft^2\ R}\right)$	$\dot{Q}_{max}$ $\left(\frac{BTU}{hr}\right)$	NTU	C_r	ϵ	$\dot{Q}$ $\left(\frac{BTU}{hr}\right)$	T_2 (°F)	T_4 (°F)
400	1798	1.16	2981	0.0048	6.0	431	4876	4053	4886	4.9	56085	0.24	0.09	0.21	11713	87	188
500	2247	1.45	3726	0.0045	7.0	539	4876	4053	4886	5.6	70106	0.22	0.11	0.19	13487	85	187
600	2696	1.74	4471	0.0042	7.9	647	4876	4053	4886	6.1	84128	0.20	0.13	0.18	14807	83	187
700	3146	2.03	5217	0.0038	8.3	755	4876	4053	4886	6.4	98149	0.18	0.15	0.16	15663	81	187
800	3595	2.32	5962	0.0035	8.7	863	4876	4053	4886	6.6	112170	0.16	0.18	0.15	16270	79	187

Table 14: Effectiveness-NTU Analysis Results - B-25 Unit Heater with Increasing Air Flow Rate

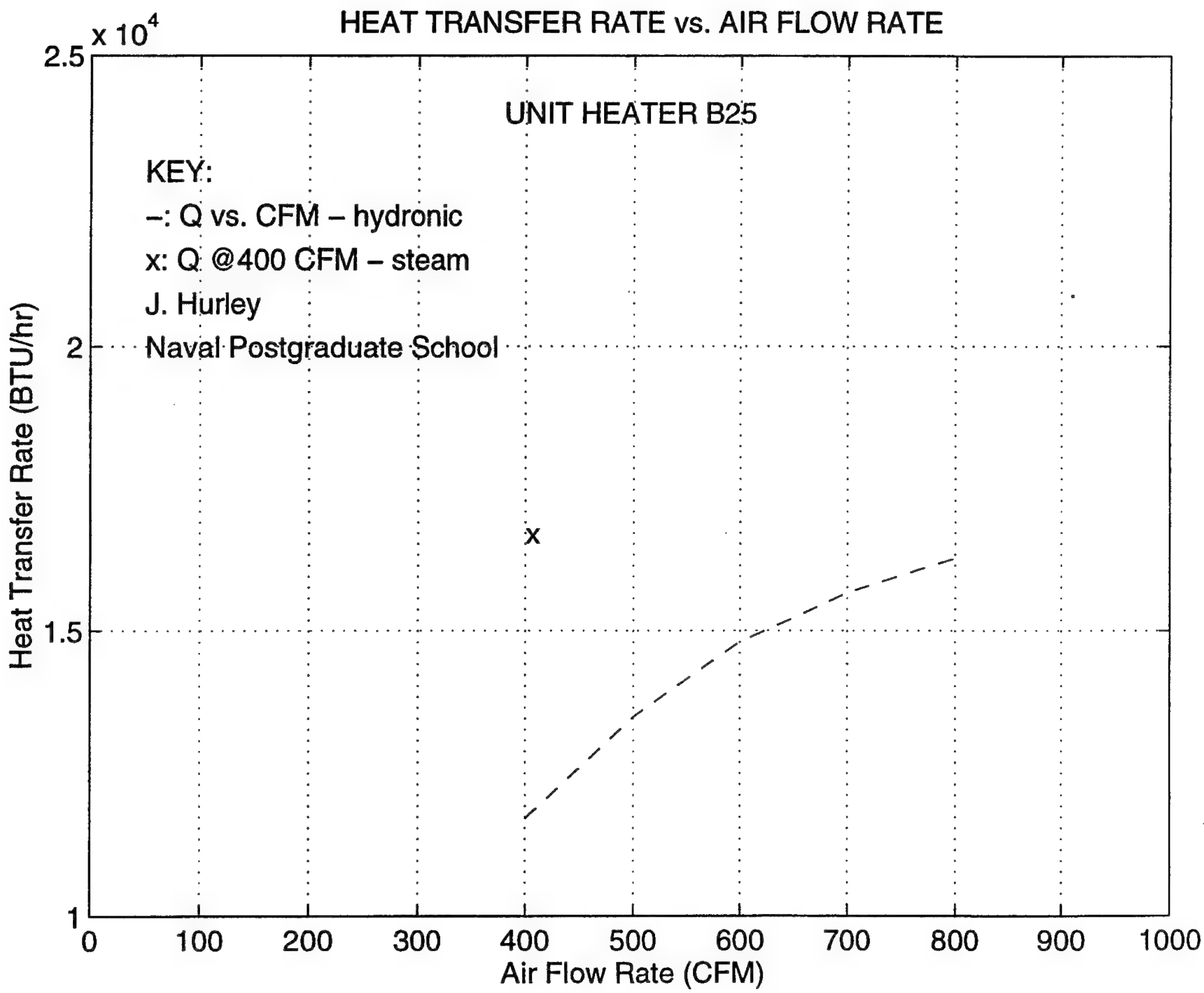


Figure 17. Heat Transfer Rate vs. Air Flow Rate - Unit Heater B25

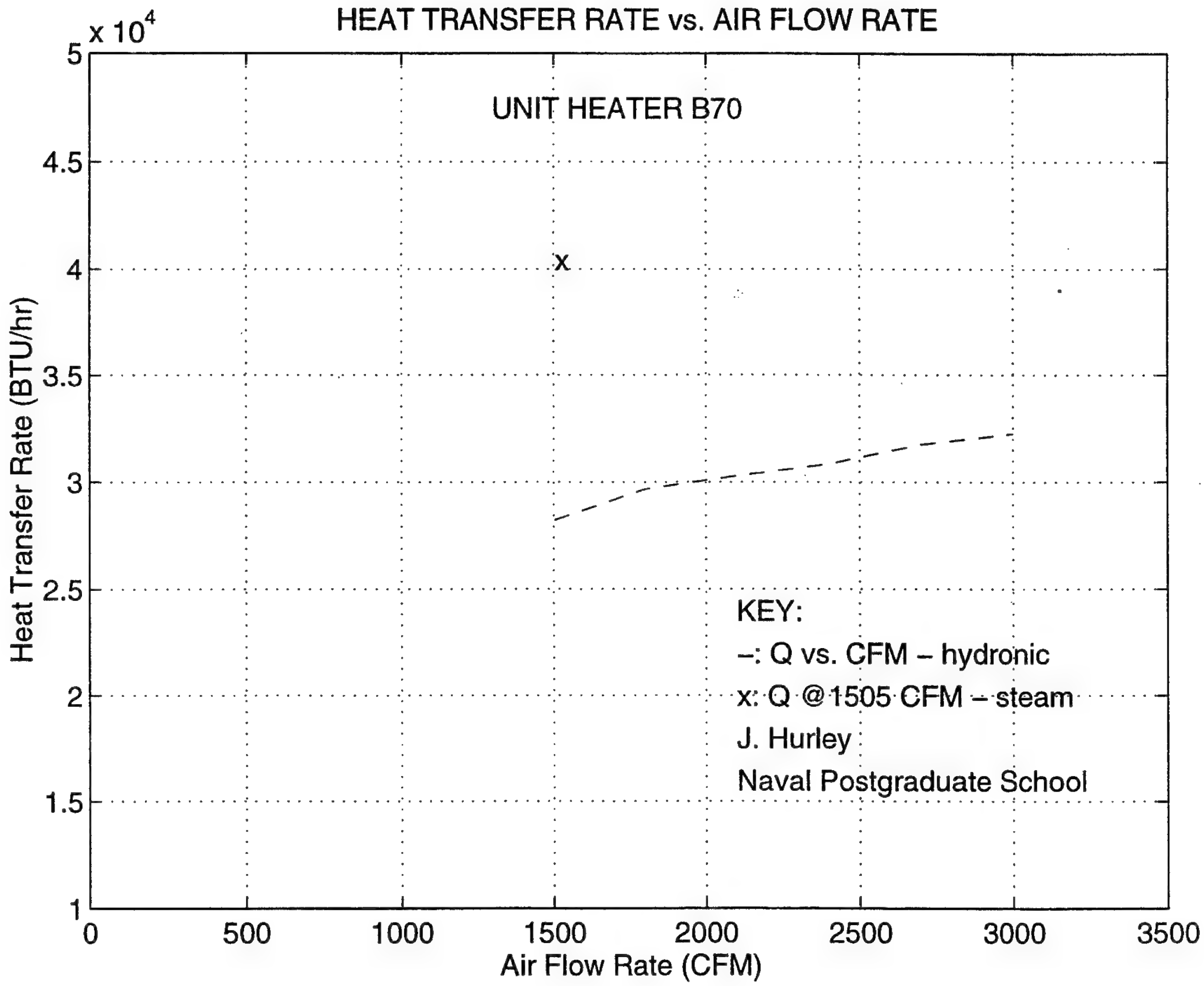


Figure 18. Heat Transfer Rate vs. Air Flow Rate - Unit Heater B70

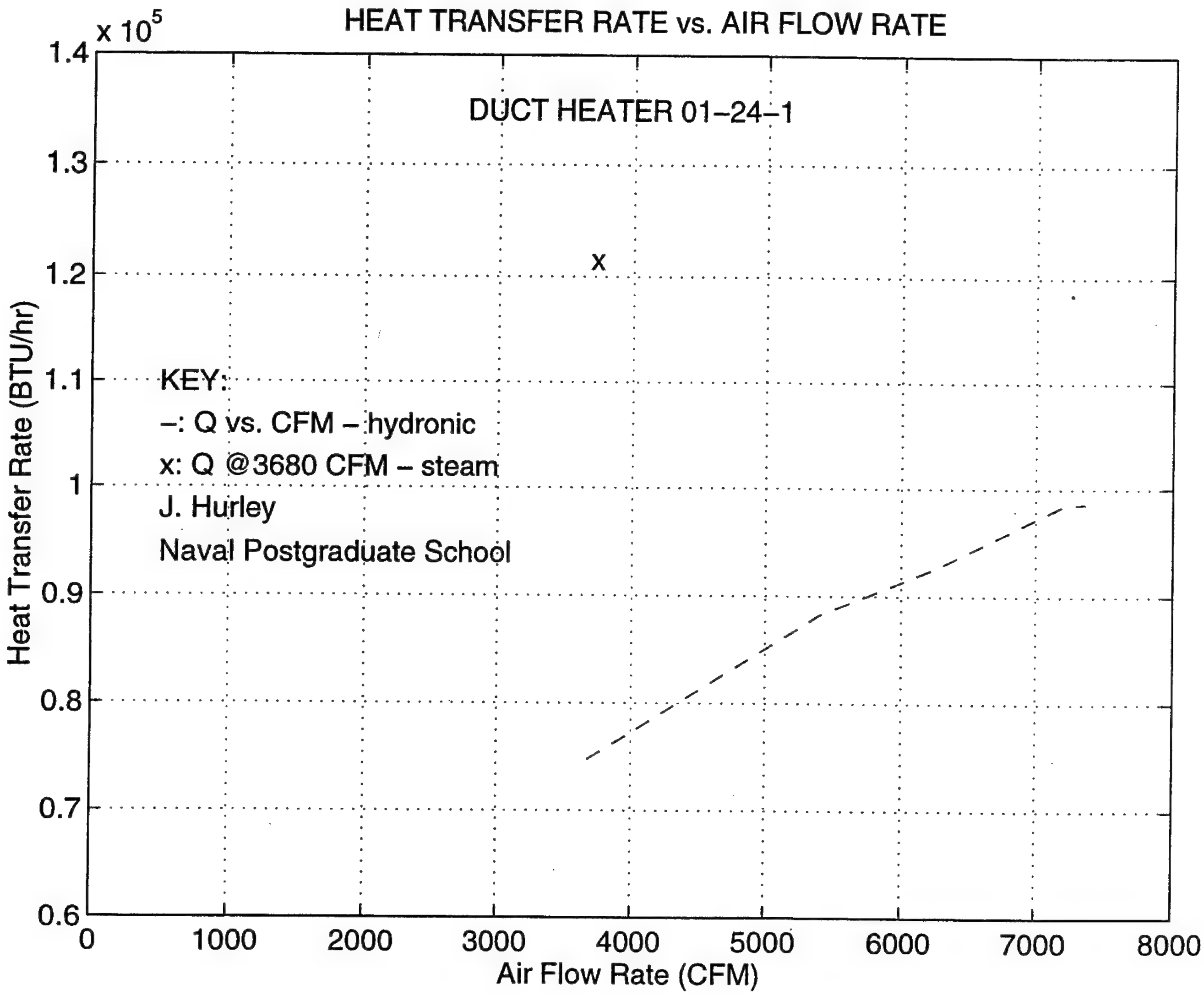


Figure 19. Heat Transfer Rate vs. Air Flow Rate - Duct Heater 01-24-1

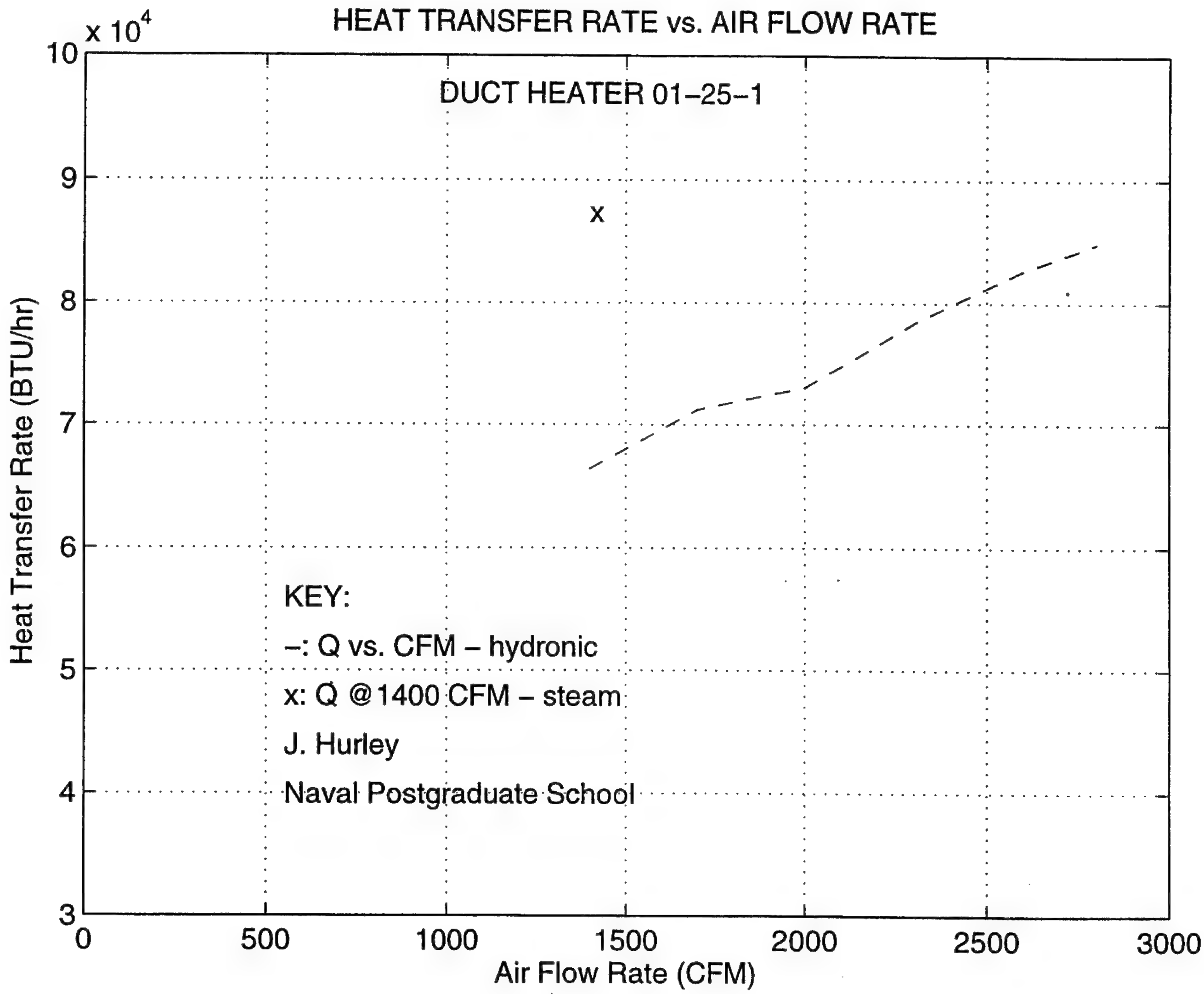


Figure 20. Heat Transfer Rate vs. Air Flow Rate - Duct Heater 01-25-1

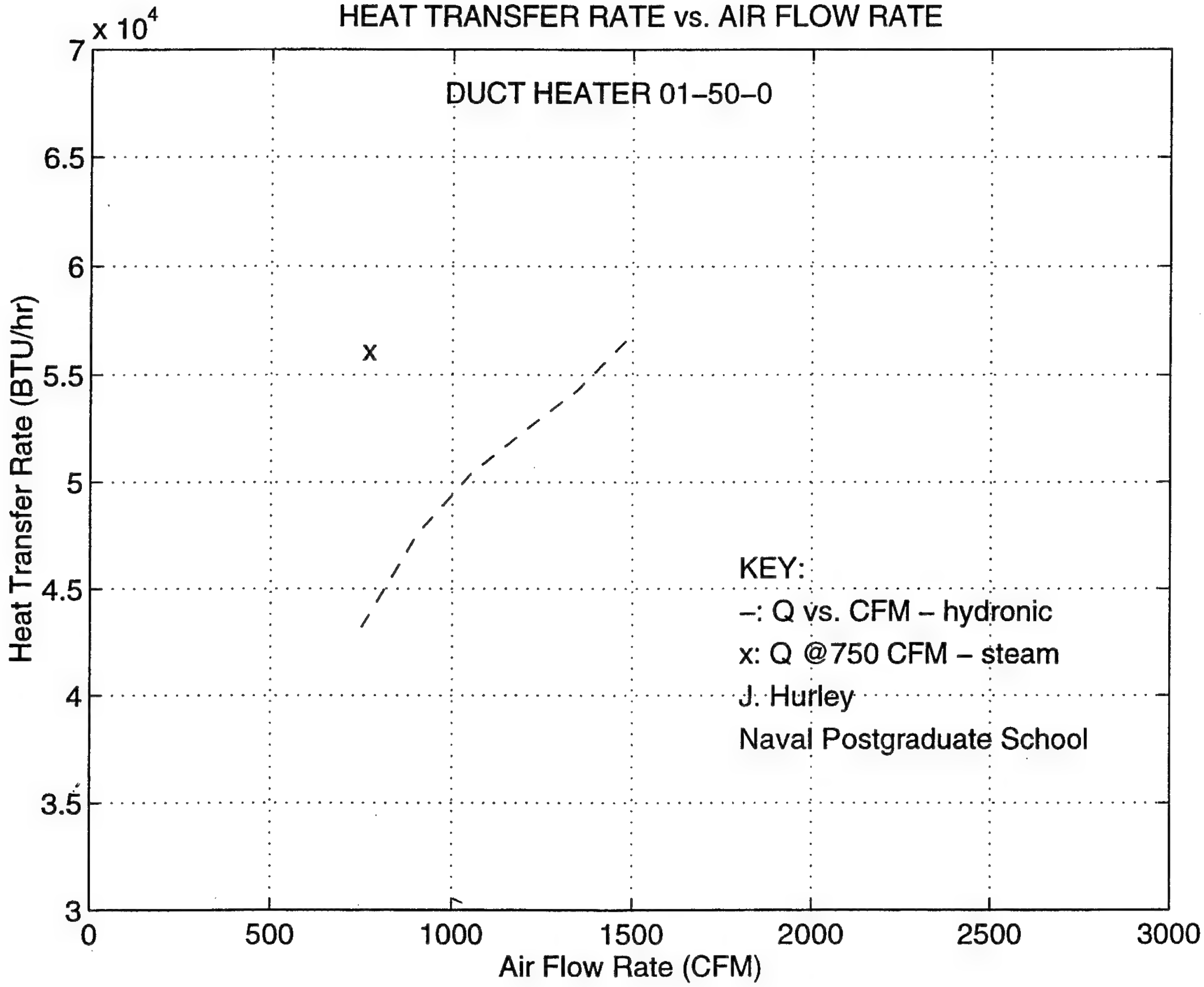


Figure 21. Heat Transfer Rate vs. Air Flow Rate - Duct Heater 01-50-0

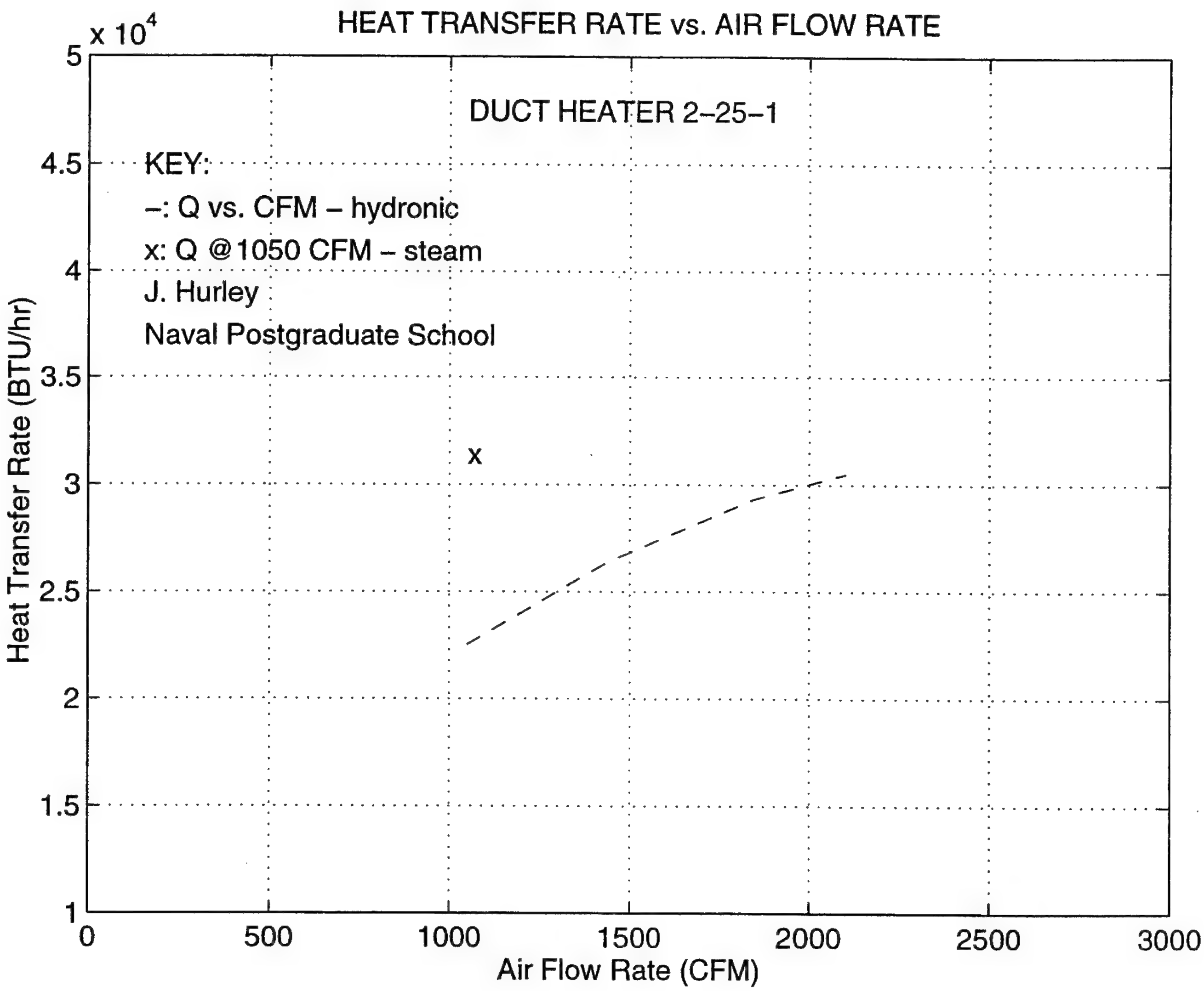


Figure 22. Heat Transfer Rate vs. Air Flow Rate - Duct Heater 2-25-1

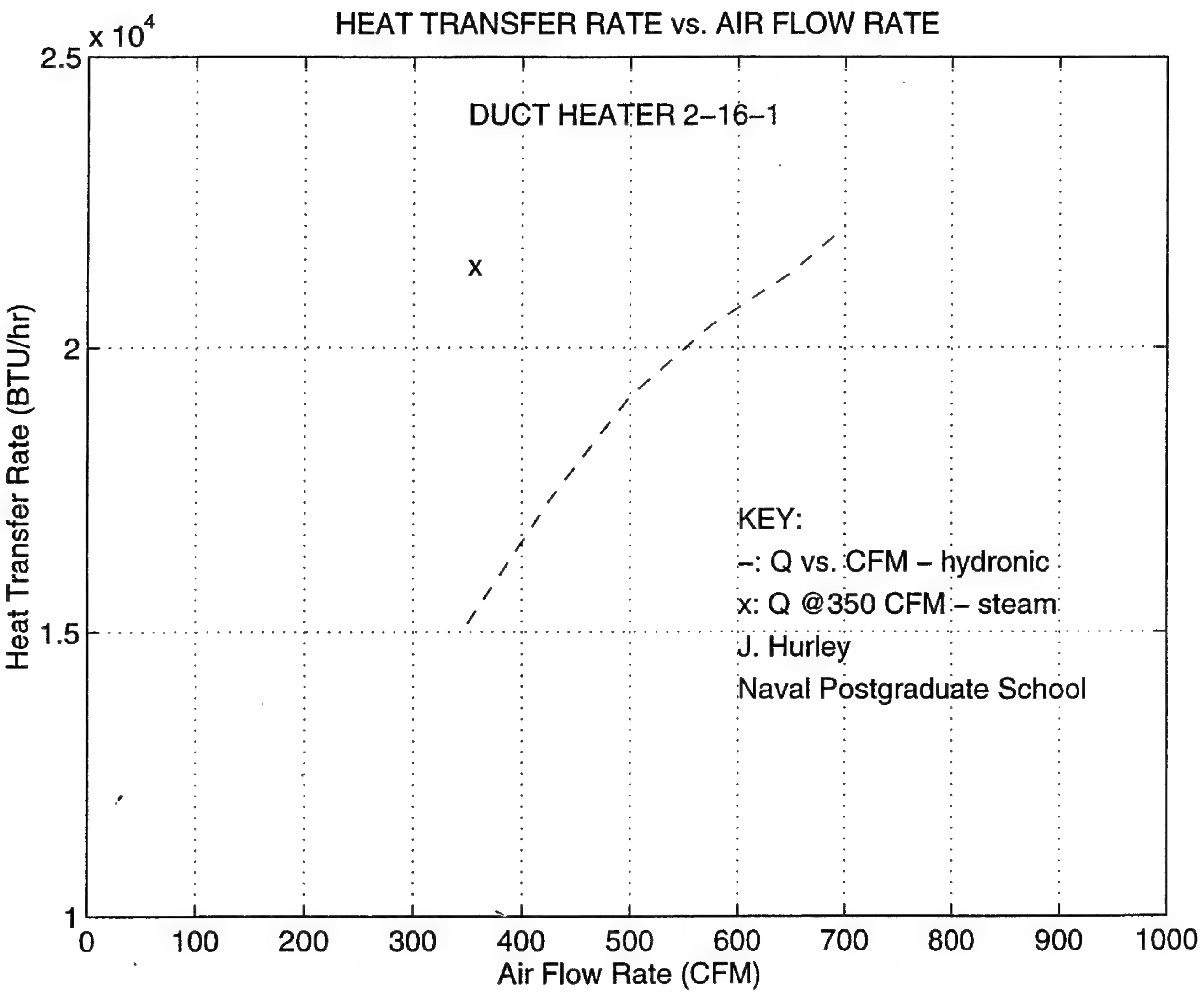


Figure 23. Heat Transfer Rate vs. Air Flow Rate - Duct Heater 2-16-1

air's heat capacity, C_{air}, also increases with larger air flow rates, resulting in a net reduction in air temperature rise.

The reason for providing heat to a particular volume of air needs to be considered to determine whether a decrease in ΔT_{air} is acceptable at a higher $\dot{Q}$. If the purpose of providing heat to a volume of air is solely to increase the air temperature of potentially very cold air, as is the case with duct preheaters, then the decrease in ΔT_{air} at a higher $\dot{Q}$ would not be acceptable. If the purpose of providing heat to a space is to provide incident heat to surfaces in the space, with the air temperature already at reasonably comfortable level, (e.g. 60°F), as could be the case with unit heaters, then the decrease in ΔT_{air} at a higher $\dot{Q}$ would be acceptable. In the latter case, the air provided to the space at an increased $\dot{Q}$.has increased enthalpy, thereby able to increase the enthalpy of a volume of 'unheated' air to a proportionally higher value.

d. Practicality

Increasing the flow rate of air passing over the duct heaters, if involving the replacement of existing fans with larger fans, is not seemingly a practical endeavor. The expense, effort, and potential ramifications of duct fan replacement make this course of action, at an initial evaluation, not viable.

Increasing the flow rate of air passing over the unit heaters, due to much better accessibility and smaller scale of the project, is seemingly a practical endeavor. An initial review of fans available from the unit heater manufacturer, New York Blower

Company, indicates that fans with larger flow rates having similar dimensions as those presently installed are available. An excerpt from a New York Blower technical publication showing various size/flow rate fans is included in Appendix E.

2. Increasing Entering Water Temperature

a. Analysis Method

The hydronic system analysis performed thus far was done assuming an entering water temperature, T_3, of 190°F, an industry accepted hydronic system operating temperature. It should be noted that a non-pressurized water system will never match the heat transfer performance of a steam system with all other system parameters equal. This is due to the difference in the water's and steam's entering temperature, which as was shown previously, directly affects the Effectiveness-NTU computation of the maximum heat transfer rate, $\dot{Q}_{max}$ which in turn affects the heat transfer rate, $\dot{Q}$, (recall that

$\dot{Q}_{max} = C_{min}(T_3 - T_1)$ *and* $\dot{Q} = \epsilon Q_{max}$).

b. Analysis Results

The effects of increasing the inlet water temperature from 190°F to 205°F are shown in Tables 15 and 16.

c. Practicality

The means to provide an increased water temperature are available from many commercial sources. An engineering judgement needs to be made as to whether the energy put into additionally heating the water is worth the energy received in the form of

Heater	$\dot{Q}_{T_3=190°F}$ (BTU/hr)	$\dot{Q}_{T_3=205°F}$ (BTU/hr)	$\frac{\dot{Q}_{T_3=205°F}}{\dot{Q}_{T_3=190°F}}$
B-25	11,713	13,064	1.11
B-70	28,214	31,469	1.11

* Type B-25 include unit heaters 2-8-0 and 2-73-1.
** Type B-70 include unit heaters 2-80-1, 2-40-1, 2-40-2, 2-60-1, 2-60-2.

Table 15: Unit Heater Hydronic Performance Data with Increased Entering Water Temp.

Heater	$\dot{Q}_{T_3=190°F}$ (BTU/hr)	$\Delta T_{air_{T_3=190°F}}$ (°F)	$\dot{Q}_{T_3=205°F}$ (BTU/hr)	$\Delta T_{air_{T_3=205°F}}$ (°F)	$\frac{\dot{Q}_{T_3=205°F}}{\dot{Q}_{T_3=190°F}}$	$\frac{\Delta T_{air_{T_3=205°F}}}{\Delta T_{air_{T_3=190°F}}}$
01-24-1	74,778	19	83,541	21	1.11	1.11
01-25-1	66,393	44	70,920	47	1.07	1.07
01-50-0	43,215	53	46,162	57	1.07	1.07
2-25-1	22,526	20	24,779	22	1.10	1.10
2-16-1	15,152	40	16,720	44	1.10	1.10

Table 16: Duct Heater Hydronic Performance Data with Increased Entering Water Temp.

heated air. A control system that would allow water temperatures to be easily adjusted to higher temperatures in colder weather is desirable. To minimally impact a hot water heating system on the whole, local electrical booster heaters could be installed

near heat exchanger water inlets. This could be a desirable means to provide hotter water to the duct preheaters, which directly heat weather-intake air.

E. KEY FACTORS IN HEAT EXCHANGER PERFORMANCE

1. Dependence on Air Side Convection Coefficient

Results of Effectiveness-NTU analysis for both the hydronic and steam system heat exchangers are dependent on the air side convection coefficient, h_{air}. Retracing the path in obtaining heat transfer rates and air temperature rises from the Effectiveness-NTU analysis, it was discerned that the most influential characteristic of a given heat exchanger was the air side convection coefficient. Due to the its relatively small magnitude, the value of h_{air} dominated computation of the overall coefficient, U_{air}. From U_{air} was determined the number of transfer units, NTU. Larger values of NTU yielded larger efficiencies, ϵ, which when multiplied by the maximum heat transfer rate, $\dot{Q}_{\max}$, yielded increased heat transfer rates, $\dot{Q}$.

Gas-liquid heat exchanger performance is thus regarded as air side limited, with the air side convection coefficient the key parameter. The manner in which h_{air} is computed is worth revisiting to determine influencing factors. Recalling from Chapter III:

$$h_{air} = \frac{\dot{Q}}{(T_S - T_1) * A_{s_{air}}} \qquad (37)$$

In this expression, the use of T_I (inlet air temperature, the lowest temperature in the system) results in the largest difference when subtracted from T_S, thus giving the most conservative (lowest) values of h_{air}. Carrying this conservative value of h_{air} through the analysis as described above results in conservative values for heat transfer performance.

2. Manufacturer's Stated Heat Transfer Performance

Using a manufacturer's air temperature rise table, such as the on shown in Figure 2, the steam system heat transfer rate can be solved for directly (i.e. $\dot{Q} = \dot{m}c_p[T_2 - T_1]$).

Proceeding to solve for ϵ, *NTU,* U_{air}, and h_{air} (neglecting all thermal resistances but air side convective resistance for simplification, i.e. $1/UA = 1/(\eta_o hA)_{air}$) will give an indication of what T_I equivalent a manufacturer's h_{air} and resulting heat transfer rates/air temperature rises are based on.

Looking at manufacturer's information presented in Figure 2, and considering the case of a single row, 12 fin per inch heat exchanger, with an air flow velocity of 500 FPM, the temperature rise from 0°F is 77.2°F. Solving for h_{air} and then the T_I equivalent as outlined above yields a value of T_I equal to 58°F (see Appendix F for computation details). This is in contrast to the value of 0°F used for computations of h_{air} in the analysis done in Chapter III (see output for Fortran computer code "hxair" I Appendix B). This demonstrates that values of h_{air} that manufacturer's heat transfer rates are based on are significantly higher than values that were used in this thesis, leading to differences in heat exchanger performance values.

The difference in the analysis starting point of T_I outlined above indicates that *magnitudes* of heat exchanger performance data found in this thesis are conservative by approximately 15-20 percent when compared to manufacture's data. The *percent difference* between the hydronic and steam system heat transfer performance will however still be consistent with the results of this thesis - compared data will have larger magnitudes but with approximately the same percent differences. The only true verification of any prediction of the complex heat transfer involved in a compact gas-liquid heat exchanger is through controlled measurements under known conditions. By experimental verification, predictions to cover similar geometries are then able to be made with more certainty.

F. OPTIMIZATION OF WATER FLOW RATES

1. Optimization Candidates

Heat exchangers that meet or exceed the heat transfer rates air temperature rises as stated in the ship's prints (heat exchangers 2-8-0, 2-80-1, and 01-24-1) can be further analyzed to determine an optimal water flow rate. With the air flow rate in this view of system operation fixed at the manufacturer's/print values, the water mass flow rate can be adjusted to lesser values to obtain an optimal value at which the minimum heat delivery rate and air temperature rise can be achieved. The objective in this optimization is to achieve the minimum water pumping power required to meet stated system heat transfer performance parameters.

2. Effectiveness-NTU Analysis Method Readapted

The Effectiveness-NTU analysis method can be used once again with the intention of solving for the heat exchanger surface area for a given heat transfer rate. The heat exchanger surface area can be solved for using a relation presented in Chapter III, i.e.

$$A_{s_{air}} = \frac{(NTU)(C_{\min})}{U_{air}} \qquad (38)$$

where, for instances of $(\dot{m}\, c_p)_{wtr} < (\dot{m}\, c_p)_{air}$:

$$C_{\min} = (\dot{m}\, c_p)_{wtr} \qquad (39)$$

Thus, for a given heat exchanger with known air flow rate, air side convection coefficient, estimated water side coefficient, and stated heat transfer rate, the air side surface area can be computed and compared with the known value. This optimization procedure is continued until the computed surface area is less than or equal to the known surface area.

Once an optimal water mass flow rate is found, the accompanying water side convection coefficient is adjusted to match the value used in the computation of the overall coefficient. This is not too onerous of a task since as was described in the water side convection computation section of Chapter III (results shown in Appendix C), the water side convection coefficients at low water flow rates (i.e. 0.5 gpm) are already an order of magnitude above the air side convection coefficients and thus have little effect on the overall coefficient.

a. Sequence of Computations

Computations involving the optimizations of water flow rates using the Effectiveness-NTU analysis were completed with assistance of Fortran computer codes. The sequence of computations in the Fortran code titled "hxNTUOPT.f", shown as part of Appendix C, is summarized in Figure 24. Program inputs are the previously computed estimated values of U_{air}, water inlet and outlet temperatures T_1 and T_3 respectively, air volumetric flow rates, heat exchanger air side surface areas $A_{s_{air}}$, and the stated design heat rates $\dot{Q}_{prints}$.

b. Tabular Results

An example of Effectiveness-NTU water mass flow rate optimization program output for heater 2-80-1 is shown in Table 17. Program outputs for other optimized heat exchangers 2-8-0 and 01-24-1 are shown in Appendix C. The shaded last row represents the final optimized parameters.

A summary of optimized water mass flow rates for unit heaters 2-8-0, 2-80-1, and duct heater 01-24-1 are shown in Table 18.

c. Low Optimal Flow Rates

As shown in Table 18, for heat exchangers 2-8-0 and 2-80-1, the minimum water mass flow rate are very low compared to the value of 10 gpm used for other heat exchangers in the system. It is apparent that these two heat exchangers could have much less heat transfer surface area on both the water and tube sides at the expense of an

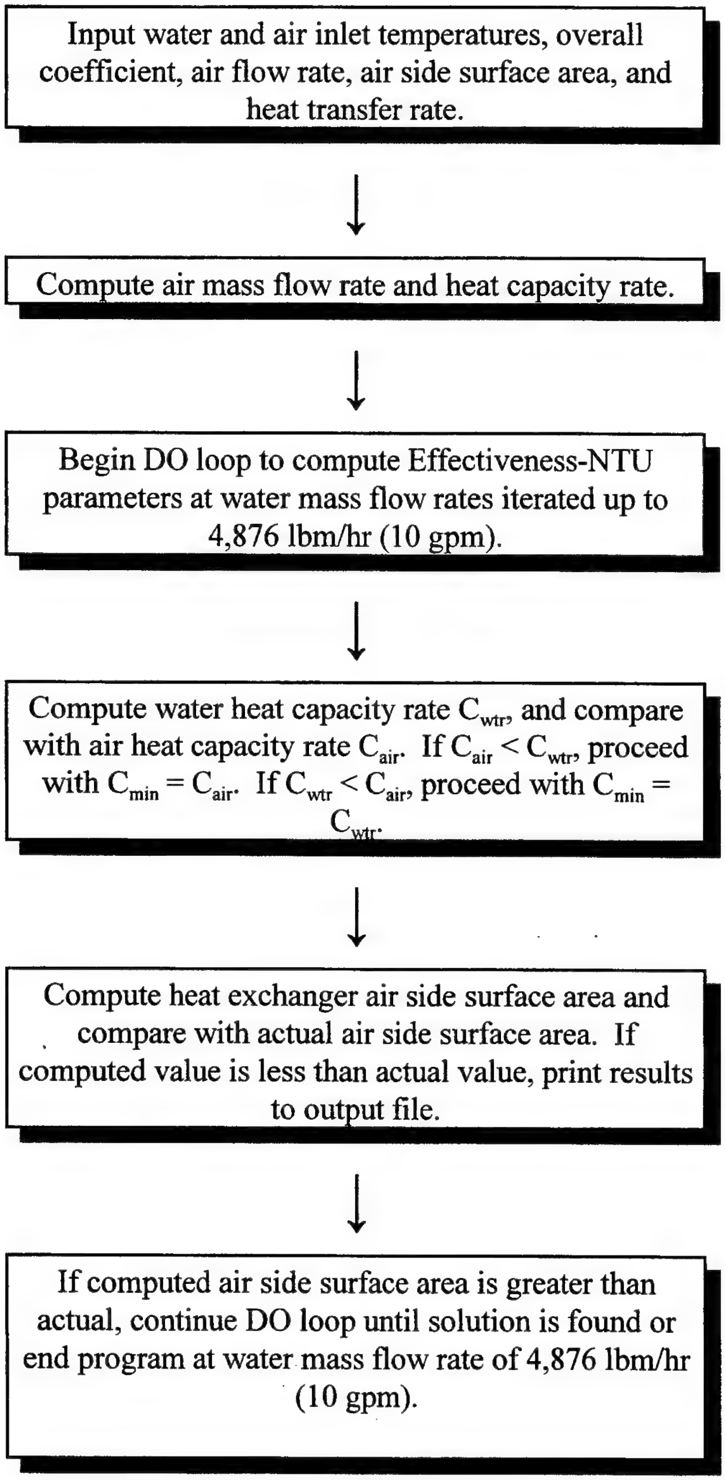


Figure 24. Effectiveness-NTU Computations Flow Chart

increased water mass flow rate and still provide heat transfer rates stated on the prints. This can be seen from the results in Table 17, where the increase in water mass flow rate corresponds to a decrease in air side surface area. Water flow rates as low as those obtained for heat exchangers 2-8-0 and 2-80-1 delivering heat transfer rates stated on the ship's prints indicates that the heat exchangers are oversized for the application. This apparent oversizing is also evident when it is noted that the same size unit heater (B-70) is used to deliver heat transfer rates varying from 5,294 to 64,860 BTU/hr. Selections of apparently oversized heat exchangers for a particular heat load was most likely based on other sound reasoning, not solely on heat transfer capacity.

$\dot{m}_{wtr}$ (lbm/hr)	$\dot{V}_{wtr}$ (gpm)	C min (BTU/hr-R)	C max (BTU/hr-R)	C_r	ϵ	NTU	Computed $A_{s_{air}}$ (in^2)
59	0.12	59.6	431.6	0.14	1.00	5.00	10,216
69	0.14	69.6	431.6	0.16	0.86	2.16	5,156
79	0.16	79.6	431.6	0.18	0.75	1.52	4,150
89	0.18	89.7	431.6	0.21	0.66	1.20	3,689
99	0.20	99.7	431.6	0.23	0.60	1.00	3,418
109	0.22	109.7	431.6	0.25	0.54	0.86	3,235
119	0.24	119.7	431.6	0.28	0.50	0.75	3,079
129	0.27	129.8	431.6	0.30	0.46	0.67	2,981

Table 17. Results of Water Mass Flow Rate Optimization Using Effectiveness-NTU Analysis for Heater 2-80-1. Actual $A_{s_{air}} = 3,013\ in^2$ and $\dot{Q}_{prints} = 7,747\ BTU/hr$.

Heater Number	$\dot{Q}_{prints}$ (BTU/hr)	Actual $A_{s_{air}}$ (in^2)	*Water Flow Rates at* $A_{s_{air}}(computed) \leq A_{s_{air}}(actual)$ (lbm/hr)	(gpm)
2-8-0	7,747	3,013	129	0.27
2-80-1	5,294	5,198	51	0.10
01-24-1	51,600	17,700	812	1.67

Table 18. Summary of Effectiveness-NTU Optimization Analysis Results

VI. CONCLUSIONS AND RECOMMENDATIONS

A. SYNOPSIS

This thesis successfully analyzed heat transfer performance aspects of a set of heat exchangers installed on the U.S. Coast Guard's WTGB Icebreaking Tug class cutters. Initial analysis with acknowledged conservative definitions of air side convection coefficients determined that the hydronic system provided on average seventy percent of the heat transfer capabilities available with the steam system. Practical improvements to the hydronic system were shown to increase heat exchanger performance parameters by an average of ten percent. It was notable that the added heat transfer available from steam is not due to a property of steam itself such as latent phase change effects, but is due solely to the increase in entering tube side temperature. Judging by heat transfer capabilities alone, with the described conservative assumptions on which these results are based, use of currently installed heat exchangers in a hydronic system is a viable option.

B. PROPOSED HYDRONIC SYSTEM

A hydronic system circulating water at 10 gallons per minute and 190 degrees F, will provide at minimum the heat transfer and air temperature rises shown in Table 10. These performance parameters were arrived at considering the entire thermal circuit involved and with conservative assumptions (when compared to manufacturer's methods) for air side convective resistance determinations. Hydronic system performance parameters can be improved through the practical methods outlined. Considering heat

transfer capabilities, use of the currently installed heat exchangers as part of a hydronic system is plausible.

Use of a pressurized water system, where system pressure would allow the circulating water temperature to equal or exceed that of steam (240°F) is another approach to the hydronic system design that could be explored.

C. FURTHER STUDY

There are numerous issues to examine further in order to take into account all aspects of using presently installed heat exchangers as part of a hydronic system. Issues remaining are related to the hardware involved as well as further analysis/design work.

One hardware issue remaining is the internal heat exchanger component's compatibilities with circulating hot water as opposed to steam. In steam heat exchangers, there is typically a strip of shaped metal which distributes the flow, insuring that the fluid passes nearly equally through all tubes. The erosion effects of water on this and other internal elements designed for use with steam should be examined.

A hardware issue on the external side of the heat exchanger is the relative location of water vs. steam inlet and outlet piping. One manufacturer's information indicated that for a steam heat exchanger, inlet piping is above the outlet to allow for condensate drainage. Conversely for circulating water, the inlet is below the outlet to help prevent air in the system from inhibiting water flow. The effort involved in possibly reversing inlet and outlet piping when switching from a steam to hydronic system thus needs to be examined.

Further design and analysis work includes among other matters, the hydronic system design to provide water at either 10 gallons per minute or the optimized values to the respective heat exchangers. Pressure drops across each heat exchanger and across other system components need to be accurately gauged to arrive at a required pumping head. Air side pressure drops also need to be examined if any of the heat transfer enhancements involving increases in air flow rate are adopted, to insure fans remain properly sized.

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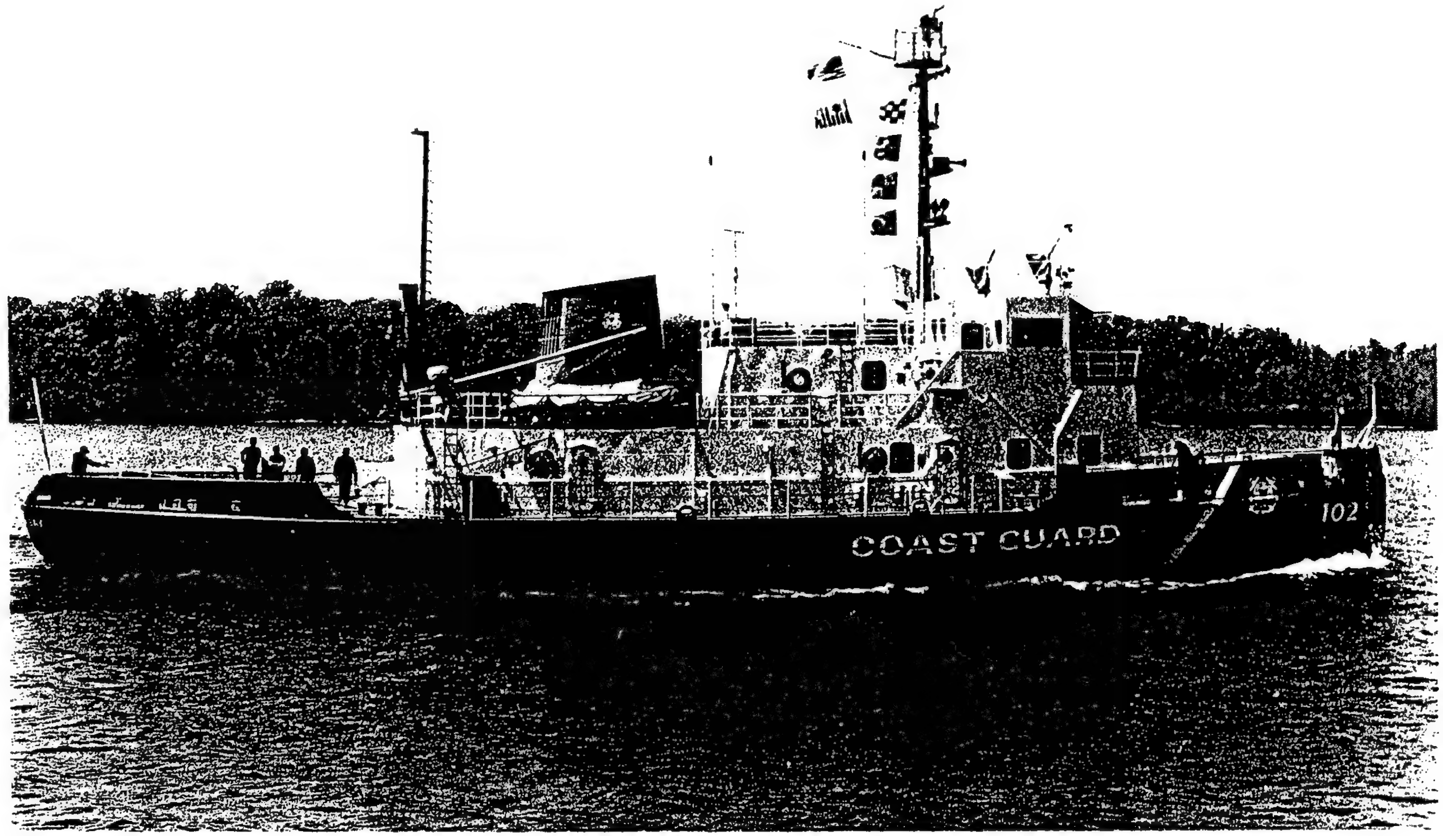
APPENDIX A. WTGB CLASS CUTTER ILLUSTRATION



United States Coast Guard Cutter Bristol Bay (WTGB-102)

APPENDIX B. HEAT EXCHANGER DIMENSIONS AND MATERIAL PROPERTIES

Heater	Coil Dimensions (in)	Fins per Inch	Frontal Area A_{fr} (in^2)	Free Flow Area A_{ff} (in^2)	Air Side Surface Area $A_{s_{air}}$ (in^2)	Fin Area to Air Side Surface Area Ratio $A_f/A_{s_{air}}$	Hydraulic Diameter D_h (in)	Area Density α (ft^2/ft^3)
B-25*	10 x 10 x 4.25	5	100	62	3,013	0.85	0.38	90
B-70*	13.5 x 13.875 x 4.125	5	185	120	5,198	0.85	0.38	83
01-24-1	20.5 x 34 x 1.88	8	693	405	17,700	0.91	0.25	163
01-25-1	12 x 24 x 1.299	12	288	155	7,959	0.96	0.154	256
01-50-0	12 x 18 x 1.299	12	216	116	5,969	0.96	0.154	256
2-25-1	12 x 16.5 x 1.299	6	198	109	2,868	0.92	0.312	134
2-16-1	12 x 12 x 1.299	12	144	77	3,979	0.96	0.154	256

* Type B-25 includes unit heaters 2-8-0 and 2-73-1.

** Type B-70 includes unit heaters 2-80-1, 2-40-1, 2-40-2, 2-60-1, 2-60-2.

Table B-1. Heat Exchanger Dimensions and Material Properties

Heater	Tube Material	Number of Tubes	Tube Dimensions (in)	Tube Wall Thickness t_{tube} (in)	Tube Thermal Conductivity k_{tube} $\frac{BTU}{hr\text{-}ft\text{-}R}$	Tube Side Surface Area $A_{s_{wtr}}$ (in^2)
B-25*	Steel	7	A = 0.31 B = 3.08	0.060	26	465
B-70*	Steel	9	A = 0.31 B = 3.08	0.060	26	807
01-24-1	Copper	12	A = 1.035 B = 1.035	0.045	221	1326
01-25-1	Copper	8	A = 0.555 B = 0.555	0.035	221	335
01-50-0	Copper	8	A = 0.555 B = 0.555	0.035	221	251
2-25-1	Copper	8	A = 0.555 B = 0.555	0.035	221	230
2-16-1	Copper	8	A = 0.555 B = 0.555	0.035	221	167

* Type B-25 includes unit heaters 2-8-0 and 2-73-1.

** Type B-70 includes unit heaters 2-80-1, 2-40-1, 2-40-2, 2-60-1, 2-60-2.

Table B-2. Heat Exchanger Dimensions and Material Properties

Heater	Fin Material	Fin Length L_{fin} (in)	Fin Width w_{fin} (in)	Fin Thickness t_{fin} (in)	Fin Thermal Conductivity k_{fin} $\frac{BTU}{hr\text{-}ft\text{-}R}$
B-25*	Steel	0.460	3.51	0.018	26
B-70*	Steel	0.460	3.51	0.018	26
01-24-1	Aluminum	0.450	1.88	0.0083	118
01-25-1	Aluminum	0.438	1.299	0.0065	118
01-50-0	Aluminum	0.438	1.299	0.0065	118
2-25-1	Aluminum	0.438	1.299	0.010	118
2-16-1	Aluminum	0.438	1.299	0.0065	118

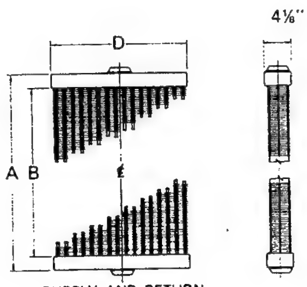
* Type B-25 includes unit heaters 2-8-0 and 2-73-1.

** Type B-70 includes unit heaters 2-80-1, 2-40-1, 2-40-2, 2-60-1, 2-60-2.

Table B-3. Heat Exchanger Dimensions and Material Properties

The
New York Blower
Company®

7660 QUINCY STREET—WILLOWBROOK, ILLINOIS 60521



SUPPLY AND RETURN
FEMALE PIPE TAP

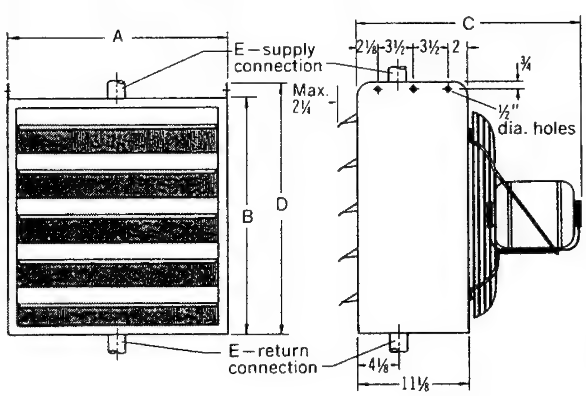
UNCASED COILS

DIMENSIONS (INCHES)

Size	Coil Face Area	A	B	D
2 *	.70	13	10	10
46	.96	13⅜	11½	11¾
812 **	1.29	15⅞	13½	13⅞
1420	1.88	19¼	16½	16¼
1824	2.78	22⅞	20	20
4256	4.92	31	27	26¼

* B-25
** B-70

DIMENSIONS [INCHES]



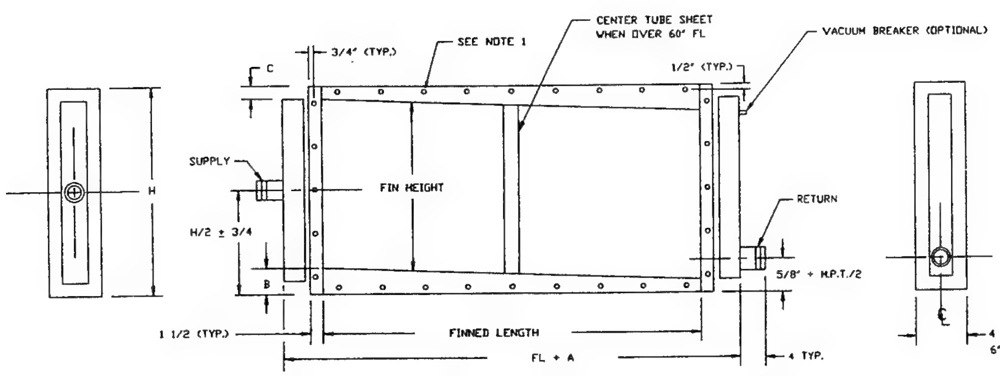
Size A or B	A	B	C max.	D	E [FPT]	Wheel dia.	Approx.* weight [lbs.]
25	12	13¼	20⅛	14¾	1½	8	65
45	13¾	13⅝	20⅛	15⅛	1½	10	75
70	15¾	16⅛	20½	17⅝	1½	12	115
105	18¼	19½	20½	21	2	14	145
120 135 155	22	23⅛	24¼	24⅝	2	18	180 180 200
200 240 270 300	28¼	31¼	24⅜	32¾	2½	24	305 305 305 310

*Weights will vary with motor specifications. Tolerance: ±⅛"

STEAM HEATING COILS 5/8" OD TUBE

Dimensions For Steam Coils

5/8" Basic, Heavy Duty Steam Heating Coil



FIN HEIGHT	TUBES HIGH	1 ROW 12-72 FIN LENGTH			1 ROW 73-120 FINNED LENGTH			2 ROW 12-72 FIN LENGTH			2 ROW 73-120 FINNED LENGTH		
		A	SUPPLY	RETURN	A	SUPPLY	RETURN	A	SUPPLY	RETURN	A	SUPPLY	RETURN
9-18	6-12	7	1 1/4 MPT	1 MPT	7 3/4	1 1/2 MPT	1 1/4 MPT	10 1/4	1 1/2 MPT	1 1/4 MPT	10 1/4	2 MPT	1 1/2 MPT
21-30	14-21	7 3/4	1 1/2 "	1 1/4 "	8 3/4	2 "	1 1/2 "	10 1/4	2 "	1 1/2 "	10 3/4	2 1/2 "	2 "
33-45	22-30	8 3/4	2 "	1 1/2 "	9 3/4	2 1/2 "	2 "	10 3/4	2 1/2 "	2 "	11 1/4	3 "	2 "

FINNED LENGTH	DIM'S B PLUS C
12" - 30"	3 1/8
31" - 60"	3 3/4
61" - 102"	4 3/4
103" - 120"	5

NOTE:
1. 5/16" DIA HOLES ON 3" CTRS FROM CENTERLINE OF CASING.
2. COIL PITCHED IN CASING TOWARD RETURN END 1/4" PER FOOT OF FINNED LENGTH. B DIM PLUS C DIM PLUS FIN HEIGHT EQUALS H.

Carrier

UNIT SIZE 39B		040	050	060
AIR QUANTITY (Cfm)		fpm	fpm	fpm
Face Velocity (fpm)	400	861	1255	2142
	500	1076	1568	2677
	600	1291	1882	3213
	700*	1507	2196	3749
Face Velocity (fpm) (H & V units only)	800	1720	2512	4288
	900	1935	2826	4824
	1000	2150	3140	5360
	1100	2365	3454	5896
	1200	2580	3768	6432
HEATING COILS Face Area (sq ft)				
U-Bend		2.15	3.14	5.36
Steam Distributing Tube		1.73	2.72	4.81
HUMIDIFIER CONN. (no. ... size)				
Atomizing Spray		1...½	1...½	1...½
Steam Grid Supply (in. OD)		1...1⅛	1...1⅛	1...1⅛
Drain (in. OD)		1...½	1...½	1...½
OPERATING WT (lb)†		450	500	750
FACE AND BYPASS DAMPER AREAS (sq ft)		6.0	7.1	11.0
MIXING BOX DAMPER AREAS (sq ft)		5.9	6.9	10.4
FANS (no. wheels...no. outlets)		1...1	1...1	1...1
Wheel	Diam** (in.)	9½	10⅝	12⅝
	Length (in.)	7⅝	8	9½
	No. of Blades	43	48	43
Shaft Critical Speed (rpm)		2700	2550	1700
Max Operating Speed (rpm)		2160	2040	1360
Fan Sheave Bore (in.)		1 3/16	1 3/16	1 3/16
Max Brake Hp		2	3	5

BASE UNIT AND ACCESSORY DIMENSIONS (in.)

BASE UNIT		040 Type A	040 Type B	040 Type D	040 H & V	050 Type A	050 Type B	050 Type D	050 H & V	060 Type A	060 Type B	060 Type D	060 H & V
FAN & COIL SECTION	L	46⅜	34⅜	19⅜	40¼	53⅜	39⅜	22⅜	33⅞	67⅞	49 7/16	28⅜	42
	W	38½	38½	38½	42⅛	38½	38½	38½	42⅛	46⅛	46⅛	46⅛	49½
	H	34¼	34¼	51¼	20¼	36⅞	36⅞	57⅞	23¾	43	43	70	29¾
MIXING BOX SECTION	L	21¼	21¼	21¼	21¼	24¾	24¾	24¾	24¾	30¾	30¾	30¾	30¾
	W	36⅝	36⅝	36⅝	36⅝	36⅝	36⅝	36⅝	36⅝	44¼	44¼	44¼	44¼
	H	25⅝	25⅝	25⅝	25⅝	29⅜	29⅜	29⅜	29⅜	37⅜	37⅜	37⅜	37⅜
LOW/HIGH VELOCITY FILTER SECTION	L	4½	4½	4½	4½	4½	4½	4½	4½	4½	4½	4½	4½
	W	36⅝	36⅝	36⅝	36⅝	36⅝	36⅝	36⅝	36⅝	44¼	44¼	44¼	44¼
	H	20¼	20¼	20¼	20¼	23¾	23¾	23¾	23¾	29¾	29¾	29¾	29¾
PREHEAT SECTION	L	6	6	6	–	7	7	7	–	9	9	9	–
	W	36⅝	36⅝	36⅝	–	36⅝	36⅝	36⅝	–	44¼	44¼	44¼	–
	H	20¼	20¼	20¼	–	23¾	23¾	23¾	–	29¾	29¾	29¾	–
BYPASS PLENUM FACE AND BYPASS DAMPER SECTION	L	6	6	6	–	7	7	7	–	9	9	9	–
	W	36⅝	36⅝	36⅝	–	36⅝	36⅝	36⅝	–	44¼	44¼	44¼	–
	H	28⅜	28⅜	28⅜	–	32⅞	32⅞	32⅞	–	41⅞	41⅞	41⅞	–
SERVICE AREA REQUIREMENTS	A	46⅜	34⅜	19⅜	40¼	53⅜	39⅜	22⅜	33⅞	67⅞	49 7/16	28⅜	42
	B	48½	48½	48½	48½	48½	48½	48½	48½	56½	56½	56½	56½

L – Length
W – Width
H – Height
H & V – Heating and Ventilating

APPENDIX C. FORTRAN COMPUTER CODES AND OUTPUTS

Jun 6 1996 12:15:03 **hxair.f** Page 1

```fortran
C
C                                                   Jim Hurley
C                                                   Naval Postgraduate School
C                                                   Spring 1996
C
      PROGRAM HEAT EXCHANGER MODEL AIR SIDE
C
      REAL CPAIR,TSTD,T1,TS,AFR,AFF,ASAIR,DH,VAIRL,VAIRU,VAIR,
     +T2(200:1200),TM,MDOTAIR,QDOTAIR,HAIR,G,RE,PR,ST,JH,DAIR,
     +MUAIR,KAIR,PATM,R,MU0,T0,S,C(0:6),N,VAIRINC
      DATA CPAIR,TSTD/0.24,70.0/
C
C     Input heat exchanger dimensional characteristics, air flow inlet
C     temperature, air velocities, and array of corresponding outlet
C     air temperatures.
C
      OPEN(55, FILE = 'RESULTS.HXAIR')
      WRITE(*,4)
      WRITE(*,*)"Air flow entering temperature (degrees F):"
      READ *,T1
      WRITE(*,*)"Heat exchanger surface temperature (degrees F):"
      READ *,TS
      WRITE(*,*)"Heat exchanger frontal area (square inches):"
      READ *,AFR
      WRITE(*,*)"Heat exchanger free flow area (square inches):"
      READ *,AFF
      WRITE(*,*)"Total air side surface area (square inches):"
      READ *,ASAIR
      WRITE(*,*)"Flow passage hydraulic diameter (inches):"
      READ *,DH
      WRITE(*,*)"Coil face velocity lower value (FPM):"
      READ *,VAIRL
      WRITE(*,*)"Coil face velocity upper value (FPM):"
      READ *,VAIRU
      WRITE(*,*)"Coil face velocity increment (FPM):"
      READ *,VAIRINC
      DO 20 VAIR = VAIRL, VAIRU, VAIRINC
        WRITE(*,5)T1,VAIR
        READ *,T2(VAIR)
   20 CONTINUE
      WRITE(55,6)T1,TS,AFR,AFF,ASAIR,DH
C
C     Begin DO loop to compute flow parameters at each flow velocity.
C

      DO 21 VAIR = VAIRL, VAIRU, VAIRINC
C
C       Compute mean air temperature to determine transport
C       properties with FUNCTION subroutines.
C
        TM = (T2(VAIR)+T1)/2.0
C
C       Using relations defined in Section III-D-1, compute parameters
C       MDOTAIR, QDOTAIR, HAIR, G, RE, PR, ST, and JH.
C
        MDOTAIR = DAIR(TSTD)*(AFR/144.0)*(VAIR*60.0)
        QDOTAIR = MDOTAIR*CPAIR*(T2(VAIR)-T1)
        HAIR = QDOTAIR/((TS-T1)*(ASAIR/144.0))
        G = (MDOTAIR*144.0)/(3600.0*AFF)
        RE = G*(DH/12.0)/MUAIR(TM)
        PR = CPAIR*MUAIR(TM)/KAIR(TM)
        ST = HAIR/(3600.0*G*CPAIR)
        JH = ST*PR**(2.0/3.0)
C
C       Print results to output file.
C
        WRITE(55,7)VAIR,T2(VAIR),MDOTAIR,QDOTAIR,HAIR,RE,JH
C
C     Continue DO loop to completion.
C
   21 CONTINUE
C
    4 FORMAT(//,"****HEAT EXCHANGER MODEL - AIR SIDE****",//,
```

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```fortran
     +"Enter the following parameters in the units",/,
     +"indicated:",/)
    5 FORMAT(/,"T2 at T1 = ",F5.1," F and VAIR = ",F5.0," FPM:")
    6 FORMAT(////,
     +T25,"HEAT EXCHANGER MODEL - AIR SIDE",/,
     +T25,"===============================",////,
     +T5,"Heat exchanger number:",////,
     +T8,"T1    = ",F5.1," degrees F",/,
     +T8,"TS    = ",F5.1," degrees F",/,
     +T8,"AFR   = ",F5.1," square inches",/,
     +T8,"AFF   = ",F5.1," square inches",/,
     +T8,"ASAIR = ",F7.1," square inches",/,
     +T8,"DH    = ",F4.2," inches",///////,
     +T1,"VAIR    T2   MDOTAIR    QDOTAIR      H-AIR      RE       JH",/,
     +T1,"(FPM)   (F)  (lbm/hr)  (BTU/hr) (BTU/hr sqft R)",/,
     +T1,"-------------------------------------------------------------")
    7 FORMAT(T1,F5.0,T8,F5.1,T15,F6.0,T24,F7.0,T36,F6.1,T46,F6.0,
     +T55,F6.5)
      END
C
C     ==========================================================
C
C     FUNCTION subroutines to compute temperature dependent transport
C     properties of air.
C
      FUNCTION DAIR(TSTD)
      REAL DAIR,TSTD,PATM,R
      DATA PATM,R/14.7,53.34/
      DAIR = (PATM*144.0)/(R*(TSTD+459.67))
      RETURN
      END
C
      FUNCTION MUAIR(TM)
      REAL MUAIR,TM,MU0,T0,S
      DATA MU0,T0,S/1.1491E-5,491.67,198.72/
      MUAIR = MU0*((((TM+459.67)/T0)**1.5)*((T0+S)/((TM+459.67)+S)))
      RETURN
      END
C
      FUNCTION KAIR(TM)
      REAL KAIR,TM,C(0:6),N
      DATA C(0),C(1),C(2),C(3),C(4),C(5),C(6)/
     +-2.276501E-3,1.2598485E-4,-1.4815235E-7,1.73550646E-10,
     +-1.066657E-13,2.47663035E-17,0.0/
      KAIR = 0.0
      DO 24 N = 0.0, 6.0, 1.0
        KAIR = C(N)*(((TM+459.67)*0.5556)**N) + KAIR
   24 CONTINUE
      KAIR = (KAIR*0.57782)/3600.0
      RETURN
      END .
```

May 22 1996 09:39:49 **RESULTS.HXAIRB25** Page 1

HEAT EXCHANGER MODEL - AIR SIDE

Heat exchanger number: B-25 (2-73-1 and 2-80-1)

```
T1    =   0.0 degrees F
TS    = 227.0 degrees F
AFR   = 100.0 square inches
AFF   =  61.7 square inches
ASAIR =  3013.0 square inches
DH    = 0.38 inches
```

VAIR (FPM)	T2 (F)	MDOTAIR (lbm/hr)	QDOTAIR (BTU/hr)	H-AIR (BTU/hr sqft R)	RE	JH
400.	70.3	1249.	21069.	4.4	2220.	.00506
500.	66.9	1561.	25062.	5.3	2782.	.00482
600.	63.3	1873.	28456.	6.0	3348.	.00456
700.	59.9	2185.	31416.	6.6	3917.	.00432
800.	56.7	2497.	33986.	7.2	4488.	.00409
900.	53.0	2810.	35739.	7.5	5064.	.00382
1000.	49.7	3122.	37237.	7.8	5642.	.00358
1100.	46.2	3434.	38076.	8.0	6224.	.00333
1200.	42.9	3746.	38571.	8.1	6809.	.00309

HEAT EXCHANGER MODEL - AIR SIDE

Heat exchanger number: B-70 (2-8-0, 2-40-1, 2-40-2, 2-60-1, 2-60-2)

```
T1    =   0.0 degrees F
TS    = 227.0 degrees F
AFR   = 185.0 square inches
AFF   = 120.0 square inches
ASAIR =  5198.0 square inches
DH    = 0.38 inches
```

VAIR (FPM)	T2 (F)	MDOTAIR (lbm/hr)	QDOTAIR (BTU/hr)	H-AIR (BTU/hr sqft R)	RE	JH
400.	70.3	2310.	38977.	4.8	2111.	.00571
500.	66.9	2888.	46365.	5.7	2646.	.00543
600.	63.3	3465.	52644.	6.4	3185.	.00514
700.	59.9	4043.	58119.	7.1	3726.	.00487
800.	56.7	4620.	62873.	7.7	4269.	.00461
900.	53.0	5198.	66117.	8.1	4817.	.00431
1000.	49.7	5775.	68889.	8.4	5367.	.00404
1100.	46.2	6353.	70441.	8.6	5921.	.00376
1200.	42.9	6930.	71356.	8.7	6476.	.00349

HEAT EXCHANGER MODEL - AIR SIDE

Heat exchanger number: 01-24-1

```
T1    =   0.0 degrees F
TS    = 227.0 degrees F
AFR   = 693.0 square inches
AFF   = 405.0 square inches
ASAIR = 17700.0 square inches
DH    = 0.25 inches
```

VAIR (FPM)	T2 (F)	MDOTAIR (lbm/hr)	QDOTAIR (BTU/hr)	H-AIR (BTU/hr sqft R)	RE	JH
400.	62.3	8654.	129391.	4.6	1552.	.00502
500.	56.2	10817.	145902.	5.2	1949.	.00453
600.	51.5	12981.	160440.	5.8	2348.	.00415
700.	47.8	15144.	173732.	6.2	2748.	.00385
800.	44.7	17307.	185674.	6.7	3148.	.00360
900.	42.3	19471.	197669.	7.1	3549.	.00341
1000.	39.9	21634.	207170.	7.4	3951.	.00322
1100.	38.1	23798.	217607.	7.8	4352.	.00307
1200.	36.2	25961.	225551.	8.1	4755.	.00292

Jun 6 1996 09:26:23 **RESULTS.HXAIR01251** Page 1

HEAT EXCHANGER MODEL - AIR SIDE
==============================

Heat exchanger number: 01-25-1

```
T1    =   0.0 degrees F
TS    = 227.0 degrees F
AFR   = 288.0 square inches
AFF   = 155.0 square inches
ASAIR =  7959.0 square inches
DH    = 0.15 inches
```

VAIR (FPM)	T2 (F)	MDOTAIR (lbm/hr)	QDOTAIR (BTU/hr)	H-AIR (BTU/hr sqft R)	RE	JH
200.	114.9	1798.	49587.	4.0	498.	.00785
400.	84.7	3596.	73107.	5.8	1020.	.00580
600.	69.7	5395.	90240.	7.2	1548.	.00478
800.	60.0	7193.	103575.	8.3	2080.	.00411
1000.	53.1	8991.	114580.	9.1	2614.	.00364
1200.	48.0	10789.	124290.	9.9	3150.	.00329

HEAT EXCHANGER MODEL - AIR SIDE
===============================

Heat exchanger number: 01-50-0

```
T1    =   0.0 degrees F
TS    = 227.0 degrees F
AFR   = 216.0 square inches
AFF   = 116.0 square inches
ASAIR =  5969.0 square inches
DH    = 0.15 inches
```

VAIR (FPM)	T2 (F)	MDOTAIR (lbm/hr)	QDOTAIR (BTU/hr)	H-AIR (BTU/hr sqft R)	RE	JH
200.	114.9	1349.	37190.	4.0	499.	.00784
400.	84.7	2697.	54830.	5.8	1022.	.00579
600.	69.7	4046.	67680.	7.2	1551.	.00477
800.	60.0	5395.	77681.	8.3	2084.	.00410
1000.	53.1	6743.	85935.	9.1	2620.	.00363
1200.	48.0	8092.	93218.	9.9	3157.	.00329

HEAT EXCHANGER MODEL - AIR SIDE
==============================

Heat exchanger number: 2-25-1

```
T1    =   0.0 degrees F
TS    = 227.0 degrees F
AFR   = 198.0 square inches
AFF   = 109.0 square inches
ASAIR =  2868.0 square inches
DH    = 0.31 inches
```

VAIR (FPM)	T2 (F)	MDOTAIR (lbm/hr)	QDOTAIR (BTU/hr)	H-AIR (BTU/hr sqft R)	RE	JH
200.	68.1	1236.	20205.	4.5	1023.	.00911
400.	51.1	2472.	30323.	6.7	2074.	.00684
600.	42.9	3709.	38185.	8.4	3133.	.00574
800.	37.7	4945.	44742.	9.9	4195.	.00505
1000.	34.1	6181.	50587.	11.2	5259.	.00457
1200.	31.5	7417.	56076.	12.4	6325.	.00422

HEAT EXCHANGER MODEL - AIR SIDE
===============================

Heat exchanger number: 2-16-1

```
T1    =   0.0 degrees F
TS    = 227.0 degrees F
AFR   = 144.0 square inches
AFF   =  77.0 square inches
ASAIR =  3979.0 square inches
DH    = 0.15 inches
```

VAIR (FPM)	T2 (F)	MDOTAIR (lbm/hr)	QDOTAIR (BTU/hr)	H-AIR (BTU/hr sqft R)	RE	JH
200.	114.9	899.	24793.	4.0	501.	.00781
400.	84.7	1798.	36553.	5.8	1026.	.00576
600.	69.7	2697.	45120.	7.2	1558.	.00475
800.	60.0	3596.	51788.	8.3	2093.	.00409
1000.	53.1	4495.	57290.	9.1	2631.	.00362
1200.	48.0	5395.	62145.	9.9	3171.	.00327

```fortran
C
C                                     Jim Hurley
C                                     Naval Postgraduate School
C                                     Spring 1996
C
      PROGRAM HEAT EXCHANGER MODEL WATER SIDE
C
      REAL PI,D,CP,MU,K,A,B,DH,PR,MDOTWTR,VDOTWTR,RE,F,NU,HWTR
C
      DATA PI,D,CP,MU,K
     +/3.14159,60.79,1.002,77.0E-7,0.386/
C
C     Input tube dimensions considering tube may not be circular.
C
      OPEN(55, FILE = 'RESULTS.HXWTR')
      WRITE(*,4)
      WRITE(*,*)"'A' dimension of tubing (inches):"
      WRITE(*,*)"Note that for a circular tube, 'A' = tube diameter."
      READ *,A
      WRITE(*,*)"'B' dimension of tubing (inches):"
      WRITE(*,*)"Note that for a circular tube, 'B' = tube diameter."
      READ *,B
C
C     Compute hydraulic diameter of tube.
C
      IF (A .EQ. B)DH = A
      IF (A .NE. B)DH = (4.0*A*B)/((2.0*A)+(2.0*B))
C
C     Begin DO loop to compute flow parameters at mass flow rates
C     ranging from 0 to 4,876 lbm/hr (10 gpm) in increments of 48.76 lbm/hr.
C
      PR = (CP*MU*32.174*3600.0)/K
      WRITE(55,5)A,B,DH,PR
      DO 21 MDOTWTR = 0.0, 4876.0, 48.76
C
C       Using relations defined in Section IV-B-2-b, compute parameters
C       RE, F, NU, and HWTR.
C
        VDOTWTR = MDOTWTR*7.481/(D*60.0)
        RE = (4.0*MDOTWTR*12.0)/(3600.0*PI*DH*MU*32.174)
        IF (RE .LE. 2300.0 .AND. A .EQ. B)NU = 4.36
        IF (RE .LE. 2300.0 .AND. A .NE. B)NU = 6.49
        IF (RE .GT. 2300.0)THEN
          F = (0.79*LOG(RE) - 1.64)**-2.0
          NU = ((F/8.0)*(RE-1000.0)*PR)/
     +         (1.0 + 12.7*SQRT(F/8.0)*(PR**(2.0/3.0) - 1.0))
        ENDIF
        HWTR = (NU*K*12.0)/DH
C
C       Print results to output file.
C
        WRITE(55,6)MDOTWTR,VDOTWTR,RE,NU,HWTR
C
C     Continue DO loop to completion.
C
   21 CONTINUE
C
    4 FORMAT(//,"****HEAT EXCHANGER MODEL - WATER SIDE****",//,
     +"Enter the following parameters in the units",/,
     +"indicated:",/)
    5 FORMAT(////,
     +T25,"HEAT EXCHANGER MODEL - WATER SIDE",/,
     +T25,"=================================",////,
     +T5,"Heat exchanger number:",///,
     +T8,"A  = ",F5.3," inches",/,
     +T8,"B  = ",F5.3," inches",/,
     +T8,"DH = ",F5.3," inches",/,
     +T8,"PR = ",F5.3,///,
     +T4,"    Flow Rates",/,
     +T4," Mass     Volume    Reynolds   Nusselt       HWTR",/,
     +T4,"(lbm/hr)  (gal/min)   Number    Number   (BTU/hr sqft R)",/,
     +T4,"--------------------------------------------------------",/)
    6 FORMAT(T4,F6.1,T15,F6.3,T24,F8.1,T36,F6.2,T49,F6.1)
```



```fortran
      END
```

HEAT EXCHANGER MODEL - WATER SIDE
=================================

Heat exchanger number: B-25 (2-73-1 and 2-80-1)
B-70 (2-8-0, 2-40-1, 2-40-2, 2-60-1, 2-60-2)

A = 0.310 inches
B = 3.080 inches
DH = 0.563 inches
PR = 2.315

Flow Rates Mass (lbm/hr)	Volume (gal/min)	Reynolds Number	Nusselt Number	HWTR (BTU/hr sqft R)
0.0	0.000	0.0	6.49	53.4
48.8	0.100	1482.9	6.49	53.4
97.5	0.200	2965.8	15.13	124.4
146.3	0.300	4448.7	23.88	196.4
195.0	0.400	5931.6	31.80	261.5
243.8	0.500	7414.5	39.21	322.4
292.6	0.600	8897.5	46.26	380.4
341.3	0.700	10380.4	53.04	436.1
390.1	0.800	11863.3	59.61	490.1
438.8	0.900	13346.2	66.00	542.7
487.6	1.000	14829.1	72.24	594.0
536.4	1.100	16312.0	78.35	644.3
585.1	1.200	17794.9	84.35	693.6
633.9	1.300	19277.8	90.25	742.1
682.6	1.400	20760.7	96.06	789.9
731.4	1.500	22243.6	101.80	837.1
780.2	1.600	23726.5	107.45	883.6
828.9	1.700	25209.4	113.04	929.6
877.7	1.800	26692.4	118.57	975.0
926.4	1.900	28175.3	124.05	1020.0
975.2	2.000	29658.2	129.47	1064.6
1024.0	2.100	31141.1	134.83	1108.7
1072.7	2.200	32624.0	140.16	1152.5
1121.5	2.300	34106.9	145.43	1195.9
1170.2	2.400	35589.8	150.67	1239.0
1219.0	2.500	37072.7	155.87	1281.7
1267.8	2.600	38555.6	161.03	1324.1
1316.5	2.700	40038.5	166.15	1366.3
1365.3	2.800	41521.4	171.25	1408.1
1414.0	2.900	43004.4	176.31	1449.7
1462.8	3.000	44487.3	181.33	1491.1
1511.6	3.100	45970.2	186.33	1532.2
1560.3	3.200	47453.1	191.30	1573.1
1609.1	3.300	48936.0	196.25	1613.7
1657.8	3.400	50418.9	201.17	1654.2
1706.6	3.500	51901.8	206.06	1694.4
1755.4	3.600	53384.7	210.93	1734.5
1804.1	3.700	54867.6	215.78	1774.3
1852.9	3.800	56350.5	220.60	1814.0
1901.6	3.900	57833.4	225.40	1853.5
1950.4	4.000	59316.3	230.18	1892.8
1999.2	4.100	60799.3	234.95	1931.9
2047.9	4.200	62282.2	239.69	1970.9
2096.7	4.300	63765.1	244.41	2009.8
2145.4	4.400	65248.0	249.11	2048.4
2194.2	4.500	66730.9	253.80	2087.0
2243.0	4.600	68213.8	258.47	2125.4
2291.7	4.700	69696.7	263.12	2163.6
2340.5	4.800	71179.6	267.75	2201.7

2389.2	4.900	72662.5	272.37	2239.7
2438.0	5.000	74145.4	276.98	2277.6
2486.8	5.100	75628.3	281.56	2315.3
2535.5	5.200	77111.2	286.14	2352.9
2584.3	5.300	78594.2	290.70	2390.4
2633.0	5.400	80077.1	295.24	2427.7
2681.8	5.501	81560.0	299.77	2465.0
2730.6	5.601	83042.9	304.29	2502.1
2779.3	5.701	84525.8	308.79	2539.2
2828.1	5.801	86008.7	313.28	2576.1
2876.8	5.901	87491.6	317.76	2612.9
2925.6	6.001	88974.5	322.23	2649.7
2974.4	6.101	90457.4	326.68	2686.3
3023.1	6.201	91940.3	331.12	2722.8
3071.9	6.301	93423.2	335.55	2759.2
3120.6	6.401	94906.1	339.97	2795.6
3169.4	6.501	96389.1	344.38	2831.8
3218.2	6.601	97872.0	348.78	2868.0
3266.9	6.701	99354.9	353.16	2904.0
3315.7	6.801	100837.8	357.54	2940.0
3364.4	6.901	102320.7	361.91	2975.9
3413.2	7.001	103803.6	366.26	3011.7
3462.0	7.101	105286.5	370.61	3047.5
3510.7	7.201	106769.4	374.94	3083.1
3559.5	7.301	108252.3	379.27	3118.7
3608.2	7.401	109735.2	383.58	3154.2
3657.0	7.501	111218.1	387.89	3189.6
3705.8	7.601	112701.1	392.19	3224.9
3754.5	7.701	114184.0	396.48	3260.2
3803.3	7.801	115666.9	400.76	3295.4
3852.0	7.901	117149.8	405.03	3330.5
3900.8	8.001	118632.7	409.29	3365.6
3949.6	8.101	120115.6	413.54	3400.5
3998.3	8.201	121598.5	417.79	3435.4
4047.1	8.301	123081.4	422.03	3470.3
4095.8	8.401	124564.3	426.26	3505.1
4144.6	8.501	126047.2	430.48	3539.8
4193.4	8.601	127530.1	434.69	3574.4
4242.1	8.701	129013.0	438.90	3609.0
4290.9	8.801	130495.9	443.10	3643.5
4339.6	8.901	131978.8	447.29	3678.0
4388.4	9.001	133461.7	451.47	3712.4
4437.2	9.101	134944.6	455.65	3746.7
4485.9	9.201	136427.5	459.82	3781.0
4534.7	9.301	137910.4	463.98	3815.2
4583.4	9.401	139393.3	468.13	3849.4
4632.2	9.501	140876.2	472.28	3883.5
4681.0	9.601	142359.1	476.42	3917.6
4729.7	9.701	143842.0	480.56	3951.6
4778.5	9.801	145325.0	484.68	3985.5
4827.2	9.901	146807.8	488.80	4019.4
4876.0	10.001	148290.7	492.92	4053.2

HEAT EXCHANGER MODEL - WATER SIDE

Heat exchanger number: 01-25-1, 01-50-0, 2-25-2, 2-16-1

A = 0.555 inches
B = 0.555 inches
DH = 0.555 inches
PR = 2.315

Flow Rates Mass (lbm/hr)	Volume (gal/min)	Reynolds Number	Nusselt Number	HWTR (BTU/hr sqft R)
0.0	0.000	0.0	4.36	36.4
48.8	0.100	1505.1	4.36	36.4
97.5	0.200	3010.2	15.41	128.6
146.3	0.300	4515.3	24.25	202.4
195.0	0.400	6020.4	32.26	269.2
243.8	0.500	7525.5	39.75	331.7
292.6	0.600	9030.6	46.88	391.2
341.3	0.700	10535.7	53.74	448.5
390.1	0.800	12040.8	60.38	503.9
438.8	0.900	13545.9	66.84	557.9
487.6	1.000	15051.0	73.16	610.6
536.4	1.100	16556.1	79.34	662.2
585.1	1.200	18061.1	85.42	712.9
633.9	1.300	19566.2	91.39	762.7
682.6	1.400	21071.3	97.27	811.8
731.4	1.500	22576.4	103.07	860.2
780.2	1.600	24081.5	108.80	908.0
828.9	1.700	25586.6	114.46	955.3
877.7	1.800	27091.7	120.05	1002.0
926.4	1.900	28596.8	125.59	1048.2
975.2	2.000	30101.9	131.08	1094.0
1024.0	2.100	31607.0	136.51	1139.3
1072.7	2.200	33112.1	141.90	1184.3
1121.5	2.300	34617.2	147.24	1228.9
1170.2	2.400	36122.3	152.54	1273.1
1219.0	2.500	37627.4	157.80	1317.0
1267.8	2.600	39132.5	163.03	1360.6
1316.5	2.700	40637.6	168.22	1403.9
1365.3	2.800	42142.7	173.37	1446.9
1414.0	2.900	43647.8	178.49	1489.7
1462.8	3.000	45152.9	183.58	1532.2
1511.6	3.100	46658.0	188.64	1574.4
1560.3	3.200	48163.1	193.68	1616.4
1609.1	3.300	49668.2	198.68	1658.2
1657.8	3.400	51173.3	203.66	1699.7
1706.6	3.500	52678.4	208.61	1741.1
1755.4	3.600	54183.4	213.54	1782.2
1804.1	3.700	55688.5	218.45	1823.2
1852.9	3.800	57193.6	223.33	1863.9
1901.6	3.900	58698.7	228.20	1904.5
1950.4	4.000	60203.8	233.04	1944.9
1999.2	4.100	61708.9	237.86	1985.1
2047.9	4.200	63214.0	242.66	2025.2
2096.7	4.300	64719.1	247.44	2065.1
2145.4	4.400	66224.2	252.20	2104.8
2194.2	4.500	67729.3	256.94	2144.4
2243.0	4.600	69234.4	261.67	2183.9
2291.7	4.700	70739.5	266.38	2223.2
2340.5	4.800	72244.6	271.07	2262.4
2389.2	4.900	73749.7	275.75	2301.4
2438.0	5.000	75254.8	280.41	2340.3
2486.8	5.100	76759.9	285.06	2379.1
2535.5	5.200	78265.0	289.69	2417.7
2584.3	5.300	79770.1	294.30	2456.2
2633.0	5.400	81275.2	298.90	2494.6
2681.8	5.501	82780.3	303.49	2532.9
2730.6	5.601	84285.4	308.06	2571.1
2779.3	5.701	85790.5	312.62	2609.1
2828.1	5.801	87295.6	317.17	2647.1
2876.8	5.901	88800.7	321.71	2684.9
2925.6	6.001	90305.8	326.23	2722.7
2974.4	6.101	91810.8	330.74	2760.3
3023.1	6.201	93315.9	335.23	2797.9
3071.9	6.301	94821.0	339.72	2835.3
3120.6	6.401	96326.1	344.19	2872.6
3169.4	6.501	97831.2	348.66	2909.9
3218.2	6.601	99336.3	353.11	2947.0
3266.9	6.701	100841.4	357.55	2984.1
3315.7	6.801	102346.5	361.98	3021.1
3364.4	6.901	103851.6	366.40	3058.0
3413.2	7.001	105356.7	370.81	3094.8
3462.0	7.101	106861.8	375.21	3131.5
3510.7	7.201	108366.9	379.60	3168.1
3559.5	7.301	109872.0	383.98	3204.7
3608.2	7.401	111377.1	388.35	3241.2
3657.0	7.501	112882.2	392.71	3277.6
3705.8	7.601	114387.3	397.06	3313.9
3754.5	7.701	115892.4	401.41	3350.1
3803.3	7.801	117397.5	405.74	3386.3
3852.0	7.901	118902.6	410.06	3422.4
3900.8	8.001	120407.7	414.38	3458.4
3949.6	8.101	121912.8	418.69	3494.3
3998.3	8.201	123417.9	422.99	3530.2
4047.1	8.301	124923.0	427.28	3566.0
4095.8	8.401	126428.0	431.56	3601.8
4144.6	8.501	127933.1	435.84	3637.5
4193.4	8.601	129438.2	440.10	3673.1
4242.1	8.701	130943.3	444.36	3708.6
4290.9	8.801	132448.4	448.61	3744.1
4339.6	8.901	133953.5	452.86	3779.5
4388.4	9.001	135458.6	457.09	3814.9
4437.2	9.101	136963.7	461.32	3850.2
4485.9	9.201	138468.8	465.54	3885.4
4534.7	9.301	139973.9	469.76	3920.6
4583.4	9.401	141478.9	473.96	3955.7
4632.2	9.501	142984.0	478.16	3990.7
4681.0	9.601	144489.1	482.36	4025.7
4729.7	9.701	145994.2	486.54	4060.7
4778.5	9.801	147499.3	490.72	4095.6
4827.2	9.901	149004.4	494.90	4130.4
4876.0	10.001	150509.5	499.06	4165.2

May 23 1996 11:10:59 RESULTS.HXWTRDUCT

HEAT EXCHANGER MODEL - WATER SIDE

Heat exchanger number: 01-24-1

A = 1.035 inches
B = 1.035 inches
DH = 1.035 inches
PR = 2.315

Flow Rates Mass (lbm/hr)	Volume (gal/min)	Reynolds Number	Nusselt Number	HWTR (BTU/hr sqft R)
0.0	0.000	0.0	4.36	19.5
48.8	0.100	807.1	4.36	19.5
97.5	0.200	1614.2	4.36	19.5
146.3	0.300	2421.2	11.55	51.7
195.0	0.400	3228.3	16.76	75.0
243.8	0.500	4035.4	21.55	96.4
292.6	0.600	4842.5	26.05	116.6
341.3	0.700	5649.6	30.34	135.8
390.1	0.800	6456.6	34.47	154.3
438.8	0.900	7263.7	38.47	172.2
487.6	1.000	8070.8	42.37	189.6
536.4	1.100	8877.9	46.17	206.6
585.1	1.200	9685.0	49.89	223.3
633.9	1.300	10492.0	53.54	239.6
682.6	1.400	11299.1	57.13	255.7
731.4	1.500	12106.2	60.66	271.5
780.2	1.600	12913.3	64.15	287.1
828.9	1.700	13720.4	67.58	302.5
877.7	1.800	14527.4	70.98	317.7
926.4	1.900	15334.5	74.33	332.7
975.2	2.000	16141.6	77.65	347.5
1024.0	2.100	16948.7	80.94	362.2
1072.7	2.200	17755.8	84.19	376.8
1121.5	2.300	18562.8	87.42	391.2
1170.2	2.400	19369.9	90.61	405.5
1219.0	2.500	20177.0	93.79	419.7
1267.8	2.600	20984.1	96.93	433.8
1316.5	2.700	21791.2	100.05	447.8
1365.3	2.800	22598.3	103.15	461.7
1414.0	2.900	23405.3	106.23	475.4
1462.8	3.000	24212.4	109.29	489.1
1511.6	3.100	25019.5	112.33	502.7
1560.3	3.200	25826.6	115.35	516.2
1609.1	3.300	26633.7	118.36	529.7
1657.8	3.400	27440.7	121.34	543.1
1706.6	3.500	28247.8	124.31	556.3
1755.4	3.600	29054.9	127.27	569.6
1804.1	3.700	29862.0	130.21	582.7
1852.9	3.800	30669.1	133.13	595.8
1901.6	3.900	31476.1	136.04	608.8
1950.4	4.000	32283.2	138.94	621.8
1999.2	4.100	33090.3	141.82	634.7
2047.9	4.200	33897.4	144.69	647.5
2096.7	4.300	34704.5	147.55	660.3
2145.4	4.400	35511.5	150.40	673.1
2194.2	4.500	36318.6	153.23	685.8
2243.0	4.600	37125.7	156.05	698.4
2291.7	4.700	37932.8	158.87	711.0
2340.5	4.800	38739.9	161.67	723.5
2389.2	4.900	39546.9	164.46	736.0

2438.0	5.000	40354.0	167.24	748.5
2486.8	5.100	41161.1	170.01	760.9
2535.5	5.200	41968.2	172.77	773.2
2584.3	5.300	42775.3	175.53	785.5
2633.0	5.400	43582.3	178.27	797.8
2681.8	5.501	44389.4	181.00	810.1
2730.6	5.601	45196.5	183.73	822.3
2779.3	5.701	46003.6	186.45	834.4
2828.1	5.801	46810.7	189.15	846.5
2876.8	5.901	47617.7	191.86	858.6
2925.6	6.001	48424.8	194.55	870.7
2974.4	6.101	49231.9	197.23	882.7
3023.1	6.201	50039.0	199.91	894.7
3071.9	6.301	50846.1	202.58	906.6
3120.6	6.401	51653.1	205.24	918.5
3169.4	6.501	52460.2	207.90	930.4
3218.2	6.601	53267.3	210.55	942.3
3266.9	6.701	54074.4	213.19	954.1
3315.7	6.801	54881.5	215.82	965.9
3364.4	6.901	55688.5	218.45	977.6
3413.2	7.001	56495.6	221.07	989.4
3462.0	7.101	57302.7	223.69	1001.1
3510.7	7.201	58109.8	226.30	1012.8
3559.5	7.301	58916.9	228.90	1024.4
3608.2	7.401	59723.9	231.49	1036.0
3657.0	7.501	60531.0	234.09	1047.6
3705.8	7.601	61338.1	236.67	1059.2
3754.5	7.701	62145.2	239.25	1070.7
3803.3	7.801	62952.3	241.82	1082.2
3852.0	7.901	63759.3	244.39	1093.7
3900.8	8.001	64566.4	246.95	1105.2
3949.6	8.101	65373.5	249.51	1116.6
3998.3	8.201	66180.6	252.06	1128.1
4047.1	8.301	66987.7	254.61	1139.5
4095.8	8.401	67794.8	257.15	1150.8
4144.6	8.501	68601.8	259.69	1162.2
4193.4	8.601	69408.9	262.22	1173.5
4242.1	8.701	70216.0	264.74	1184.8
4290.9	8.801	71023.1	267.27	1196.1
4339.6	8.901	71830.1	269.78	1207.4
4388.4	9.001	72637.2	272.29	1218.6
4437.2	9.101	73444.3	274.80	1229.8
4485.9	9.201	74251.4	277.30	1241.0
4534.7	9.301	75058.4	279.80	1252.2
4583.4	9.401	75865.5	282.30	1263.4
4632.2	9.501	76672.6	284.79	1274.5
4681.0	9.601	77479.7	287.27	1285.6
4729.7	9.701	78286.8	289.75	1296.7
4778.5	9.801	79093.8	292.23	1307.8
4827.2	9.901	79900.9	294.70	1318.9
4876.0	10.001	80708.0	297.17	1329.9

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C
C                                              Jim Hurley
C                                              Naval Postgraduate School
C                                              Spring 1996
C
      PROGRAM HEAT EXCHANGER OA COEFFICIENT WATER
C
      REAL HAIR(20),HWTR,KFIN,LFIN,WFIN,TFIN,ASAIR,ASWTR,ASFASAIR,
     +KTUBE,TTUBE,ATUBE,LTUBE,ODTUBE,IDTUBE,P,AX,M,LC,ETAF,ETAO,
     +UAIR,RFAIR,RFWTR,PI
      INTEGER FLAG,N,J,L
      DATA RFAIR,RFWTR,PI/0.002,0.0005,3.14159/
C
C     Input water and air side convection coefficients and heat exchanger
C     dimensional characteristics.  Classify computations as either for
C     a unit heater or a duct heater.
C
      OPEN(55, FILE = 'RESULTS.HXU')
      WRITE(*,4)
      WRITE(*,*)"Convection coefficient - water side (BTU/hr sqft R):"
      READ *,HWTR
      WRITE(*,*)"Quantity of convection coefficients - air side that"
      WRITE(*,*)"will be entered."
      READ *,N
      DO 21 J = 1, N, 1
        WRITE(*,5)J
        READ *,HAIR(J)
   21 CONTINUE
      WRITE(*,*)"Conduction coefficient - fin (BTU/hr ft R):"
      READ *,KFIN
      WRITE(*,*)"Fin length (inches):"
      READ *,LFIN
      WRITE(*,*)"Fin width (inches):"
      READ *,WFIN
      WRITE(*,*)"Fin thickness (inches):"
      READ *,TFIN
      WRITE(*,*)"Total air side surface area (sq. in.):"
      READ *,ASAIR
      WRITE(*,*)"Total water side surface area (sq. in.):"
      READ *,ASWTR
      WRITE(*,*)"Fin surface area/total air side surface area:"
      READ *,ASFASAIR
C
      WRITE(*,*)"Enter '1' if unit heater or '2' if duct heater:"
      READ *, FLAG
      IF (FLAG .EQ. 1)THEN
        WRITE(*,*)"Conduction coefficient - tube (BTU/hr ft R):"
        READ *,KTUBE
        WRITE(*,*)"Tube thickness (inches):"
        READ *,TTUBE
        WRITE(*,*)"Water side surface area of one tube (sq. in.):"
        READ *,ATUBE
        WRITE(55,6)HWTR,KFIN,LFIN,WFIN,TFIN,ASAIR,ASWTR,ASFASAIR,
     +  KTUBE,TTUBE,ATUBE,RFAIR,RFWTR
      ENDIF
      IF(FLAG .EQ. 2)THEN
        WRITE(*,*)"Conduction coefficient - tube (BTU/hr ft R):"
        READ *,KTUBE
        WRITE(*,*)"Length of one tube (inches):"
        READ *,LTUBE
        WRITE(*,*)"Tube outer diameter (inches):"
        READ *,ODTUBE
        WRITE(*,*)"Tube inner diameter (inches):"
        READ *,IDTUBE
        WRITE(55,7)HWTR,KFIN,LFIN,WFIN,TFIN,ASAIR,ASWTR,ASFASAIR,
     +  KTUBE,LTUBE,ODTUBE,IDTUBE,RFAIR,RFWTR
      ENDIF
C
C     Begin DO loop to compute parameters for each air side convection
C     coefficient input.
C
      DO 22 L = 1, N, 1
C
```



```
C         Using relations defined in Section IV-B-2-c, compute single fin
C         efficiency and overall efficiency.
C
          P = ((2.0*WFIN)+(2.0*TFIN))/12.0
          AX = (WFIN*TFIN)/144.0
          M = SQRT((HAIR(L)*P)/(KFIN*AX))
          LC = (LFIN+(TFIN/2.0))/12.0
          ETAF = (TANH(M*LC))/(M*LC)
          ETAO = 1.0-ASFASAIR*(1.0-ETAF)
C
C         Using relations defined in Section IV-B-3, compute overall heat
C         transfer coefficient.
C
          IF (FLAG .EQ. 1)THEN
            UAIR = 1.0/
     +      (((1.0/(ETAO*HAIR(L)*(ASAIR/144.0)))+
     +      (RFAIR/(ETAO*(ASAIR/144.0)))+
     +      (1.0/(HWTR*(ASWTR/144.0)))+(RFWTR/(ASWTR/144.0))+
     +      ((TTUBE/12.0)/(KTUBE*(ATUBE/144.0))))
     +      *(ASAIR/144.0))
          ENDIF
          IF (FLAG .EQ. 2)THEN
            UAIR = 1.0/
     +      (((1.0/(ETAO*HAIR(L)*(ASAIR/144.0)))+
     +      (RFAIR/(ETAO*(ASAIR/144.0)))+
     +      (1.0/(HWTR*(ASWTR/144.0)))+(RFWTR/(ASWTR/144.0))+
     +      (LOG(ODTUBE/IDTUBE)/(2.0*PI*(LTUBE/12.0)*KTUBE)))
     +      *(ASAIR/144.0))
          ENDIF
C
C         Print results to output file.
C
          WRITE(55,8)HAIR(L),ETAF,ETAO,UAIR
C
C       Continue DO loop to completion.
C
   22   CONTINUE
C
    4   FORMAT(//,"**HEAT EXCHANGER MODEL - OVERALL COEFFICIENT**",//,
       +"Enter the following parameters in the units",/,
       +"indicated:",/)
    5   FORMAT(" #",I2," Convection coefficient - air side",
       +" (BTU/hr sqft R):")
    6   FORMAT(////,
       +T25,"HEAT EXCHANGER OVERALL COEFFICIENT",/,
       +T25,"==================================",////,
       +T5,"Heat exchanger number:",////,
       +T8,"H-WTR      = ",F6.1," BTU/hr sqft R",/,
       +T8,"KFIN       = ",F5.1," BTU/hr ft R",/,
       +T8,"LFIN       = ",F5.3," inches",/,
       +T8,"WFIN       = ",F5.3," inches",/
       +T8,"TFIN       = ",F6.4," inches",/,
       +T8,"ASAIR      = ",F7.1," square inches",/,
       +T8,"ASWTR      = ",F6.1," square inches",/,
       +T8,"ASF/ASAIR = ",F4.2,/,
       +T8,"KTUBE      = ",F5.1," BTU/hr ft R",/,
       +T8,"TTUBE      = ",F5.3," inches",/,
       +T8,"ATUBE      = ",F5.1," square inches",/,
       +T8,"RFAIR      = ",F6.4," hr sqft R/BTU",/,
       +T8,"RFWTR      = ",F6.4," hr sqft R/BTU",///,
       +T5,"    H-AIR      ETA 1 FIN    ETA OVERALL      U-AIR",/,
       +T5,"(BTU/hr sqft R)                             (BTU/hr sqft R)",/,
       +T5,"--------------------------------------------------------")
    7   FORMAT(////,
       +T25,"HEAT EXCHANGER OVERALL COEFFICIENT",/,
       +T25,"==================================",////,
       +T5,"Heat exchanger number:",////,
       +T8,"H-WTR      = ",F6.1," BTU/hr sqft R",/,
       +T8,"KFIN       = ",F5.1," BTU/hr ft R",/,
       +T8,"LFIN       = ",F5.3," inches",/,
       +T8,"WFIN       = ",F5.3," inches",/
       +T8,"TFIN       = ",F6.4," inches",/,
       +T8,"ASAIR      = ",F7.1," square inches",/,
```

```
     +T8,"ASWTR      = ",F6.1," square inches",/,
     +T8,"ASF/ASAIR = ",F4.2,/,
     +T8,"KTUBE      = ",F5.1," BTU/hr ft R",/,
     +T8,"LTUBE      = ",F5.1," inches",/,
     +T8,"ODTUBE     = ",F5.3," inches",/,
     +T8,"IDTUBE     = ",F5.3," inches",/,
     +T8,"RFAIR      = ",F6.4," hr sqft R/BTU",/,
     +T8,"RFWTR      = ",F6.4," hr sqft R/BTU",///,
     +T5,"     H-AIR      ETA 1 FIN    ETA OVERALL      U-AIR",/,
     +T5,"(BTU/hr sqft R)                           (BTU/hr sqft R)",/,
     +T5,"-------------------------------------------------------")
    8 FORMAT(T10,F4.1,T23,F4.2,T37,F4.2,T51,F4.1)
      END
```

HEAT EXCHANGER OVERALL COEFFICIENT

Heat exchanger number: B-25 (2-73-1 and 2-80-1)

```
H-WTR     = 4053.0 BTU/hr sqft R
KFIN      =  26.0 BTU/hr ft R
LFIN      = 0.460 inches
WFIN      = 3.510 inches
TFIN      = 0.018 inches
ASAIR     =  3013.0 square inches
ASWTR     =  465.0 square inches
ASF/ASAIR = 0.85
KTUBE     =  26.0 BTU/hr ft R
TTUBE     = 0.060 inches
ATUBE     =  67.0 square inches
RFAIR     = 0.0020 hr sqft R/BTU
RFWTR     = 0.0005 hr sqft R/BTU
```

H-AIR (BTU/hr sqft R)	ETA 1 FIN	ETA OVERALL	U-AIR (BTU/hr sqft R)
6.0	0.87	0.89	4.9
7.0	0.85	0.87	5.6
7.9	0.83	0.86	6.1
8.3	0.83	0.85	6.4
8.7	0.82	0.85	6.6

HEAT EXCHANGER OVERALL COEFFICIENT

Heat exchanger number: B-70 (2-8-0, 2-40-1, 2-40-2, 2-60-1, 2-60-2)

```
H-WTR      = 4053.0 BTU/hr sqft R
KFIN       =   26.0 BTU/hr ft R
LFIN       = 0.460 inches
WFIN       = 3.510 inches
TFIN       = 0.0180 inches
ASAIR      =  5198.0 square inches
ASWTR      =  807.0 square inches
ASF/ASAIR  = 0.85
KTUBE      =   26.0 BTU/hr ft R
TTUBE      = 0.060 inches
ATUBE      =   90.0 square inches
RFAIR      = 0.0020 hr sqft R/BTU
RFWTR      = 0.0005 hr sqft R/BTU
```

H-AIR (BTU/hr sqft R)	ETA 1 FIN	ETA OVERALL	U-AIR (BTU/hr sqft R)
9.0	0.82	0.84	6.7
9.6	0.81	0.84	7.0
9.8	0.80	0.83	7.1
10.0	0.80	0.83	7.2
10.4	0.79	0.83	7.4
10.6	0.79	0.82	7.5

HEAT EXCHANGER OVERALL COEFFICIENT

Heat exchanger number: 01-24-1

```
H-WTR     = 1330.0 BTU/hr sqft R
KFIN      = 118.0 BTU/hr ft R
LFIN      = 0.450 inches
WFIN      = 1.880 inches
TFIN      = 0.008 inches
ASAIR     = 17700.0 square inches
ASWTR     = 1326.0 square inches
ASF/ASAIR = 0.91
KTUBE     = 221.0 BTU/hr ft R
LTUBE     =  34.0 inches
ODTUBE    = 1.125 inches
IDTUBE    = 1.035 inches
RFAIR     = 0.0020 hr sqft R/BTU
RFWTR     = 0.0005 hr sqft R/BTU
```

H-AIR (BTU/hr sqft R)	ETA 1 FIN	ETA OVERALL	U-AIR (BTU/hr sqft R)
7.0	0.93	0.93	5.7
7.7	0.92	0.93	6.2
8.5	0.91	0.92	6.7
9.0	0.91	0.91	7.0
9.6	0.90	0.91	7.4
9.7	0.90	0.91	7.4

HEAT EXCHANGER OVERALL COEFFICIENT

Heat exchanger number: 01-25-1

```
H-WTR      = 4165.0 BTU/hr sqft R
KFIN       = 118.0 BTU/hr ft R
LFIN       = 0.438 inches
WFIN       = 1.299 inches
TFIN       = 0.0065 inches
ASAIR      =  7959.0 square inches
ASWTR      =  335.0 square inches
ASF/ASAIR  = 0.96
KTUBE      = 221.0 BTU/hr ft R
LTUBE      =  24.0 inches
ODTUBE     = 0.625 inches
IDTUBE     = 0.555 inches
RFAIR      = 0.0020 hr sqft R/BTU
RFWTR      = 0.0005 hr sqft R/BTU
```

H-AIR (BTU/hr sqft R)	ETA 1 FIN	ETA OVERALL	U-AIR (BTU/hr sqft R)
8.2	0.90	0.90	6.4
8.9	0.89	0.90	6.8
9.2	0.89	0.89	6.9
10.0	0.88	0.88	7.4
10.7	0.87	0.88	7.8
11.2	0.87	0.87	8.0

HEAT EXCHANGER OVERALL COEFFICIENT
==================================

Heat exchanger number: 01-50-0

```
H-WTR     = 4165.0 BTU/hr sqft R
KFIN      = 118.0 BTU/hr ft R
LFIN      = 0.438 inches
WFIN      = 1.299 inches
TFIN      = 0.0065 inches
ASAIR     =  5969.0 square inches
ASWTR     =  251.0 square inches
ASF/ASAIR = 0.96
KTUBE     = 221.0 BTU/hr ft R
LTUBE     =  18.0 inches
ODTUBE    = 0.625 inches
IDTUBE    = 0.555 inches
RFAIR     = 0.0020 hr sqft R/BTU
RFWTR     = 0.0005 hr sqft R/BTU
```

H-AIR (BTU/hr sqft R)	ETA 1 FIN	ETA OVERALL	U-AIR (BTU/hr sqft R)
7.0	0.91	0.92	5.6
7.7	0.90	0.91	6.1
8.2	0.90	0.90	6.4
8.6	0.89	0.90	6.6
9.0	0.89	0.89	6.8
9.5	0.88	0.89	7.1

HEAT EXCHANGER OVERALL COEFFICIENT
==================================

Heat exchanger number: 2-25-1

```
H-WTR      = 4165.0 BTU/hr sqft R
KFIN       = 118.0 BTU/hr ft R
LFIN       = 0.438 inches
WFIN       = 1.299 inches
TFIN       = 0.0100 inches
ASAIR      =  2868.0 square inches
ASWTR      =  230.0 square inches
ASF/ASAIR  = 0.92
KTUBE      = 221.0 BTU/hr ft R
LTUBE      =  16.5 inches
ODTUBE     = 0.625 inches
IDTUBE     = 0.555 inches
RFAIR      = 0.0020 hr sqft R/BTU
RFWTR      = 0.0005 hr sqft R/BTU
```

H-AIR (BTU/hr sqft R)	ETA 1 FIN	ETA OVERALL	U-AIR (BTU/hr sqft R)
10.1	0.92	0.92	8.3
11.1	0.91	0.92	9.0
12.1	0.90	0.91	9.7
12.9	0.90	0.90	10.2
13.8	0.89	0.90	10.7
14.5	0.88	0.89	11.1

HEAT EXCHANGER OVERALL COEFFICIENT

Heat exchanger number: 2-16-1

```
H-WTR      = 4165.0 BTU/hr sqft R
KFIN       = 118.0 BTU/hr ft R
LFIN       = 0.438 inches
WFIN       = 1.299 inches
TFIN       = 0.0065 inches
ASAIR      =  3979.0 square inches
ASWTR      =  167.0 square inches
ASF/ASAIR  = 0.96
KTUBE      = 221.0 BTU/hr ft R
LTUBE      =  12.0 inches
ODTUBE     = 0.625 inches
IDTUBE     = 0.555 inches
RFAIR      = 0.0020 hr sqft R/BTU
RFWTR      = 0.0005 hr sqft R/BTU
```

H-AIR (BTU/hr sqft R)	ETA 1 FIN	ETA OVERALL	U-AIR (BTU/hr sqft R)
5.4	0.93	0.93	4.5
6.2	0.92	0.92	5.1
7.0	0.91	0.92	5.6
7.4	0.91	0.91	5.9
7.8	0.90	0.91	6.1
8.1	0.90	0.90	6.3

```
C
C                                          Jim Hurley
C                                          Naval Postgraduate School
C                                          Spring 1996
C
      PROGRAM HEAT EXCHANGER OA COEFFICIENT STEAM
C
      REAL HAIR(20),KFIN,LFIN,WFIN,TFIN,ASAIR,ASWTR,ASFASAIR,
     +KTUBE,TTUBE,ATUBE,LTUBE,ODTUBE,IDTUBE,P,AX,M,LC,ETAF,ETAO,
     +UAIR,RFAIR,RFWTR,PI
      INTEGER FLAG,N,J,L
      DATA RFAIR,RFWTR,PI/0.002,0.0005,3.14159/
C
C     Input water and air side convection coefficients and heat exchanger
C     dimensional characteristics.  Classify computations as either for
C     a unit heater or a duct heater.
C
      OPEN(55, FILE = 'RESULTS.HXU')
      WRITE(*,4)
      WRITE(*,*)"Quantity of convection coefficients - air side that"
      WRITE(*,*)"will be entered."
      READ *,N
      DO 21 J = 1, N, 1
        WRITE(*,5)J
        READ *,HAIR(J)
   21 CONTINUE
      WRITE(*,*)"Conduction coefficient - fin (BTU/hr ft R):"
      READ *,KFIN
      WRITE(*,*)"Fin length (inches):"
      READ *,LFIN
      WRITE(*,*)"Fin width (inches):"
      READ *,WFIN
      WRITE(*,*)"Fin thickness (inches):"
      READ *,TFIN
      WRITE(*,*)"Total air side surface area (sq. in.):"
      READ *,ASAIR
      WRITE(*,*)"Total water side surface area (sq. in.):"
      READ *,ASWTR
      WRITE(*,*)"Fin surface area/total air side surface area:"
      READ *,ASFASAIR
C
      WRITE(*,*)"Enter '1' if unit heater or '2' if duct heater:"
      READ *, FLAG
      IF (FLAG .EQ. 1)THEN
        WRITE(*,*)"Conduction coefficient - tube (BTU/hr ft R):"
        READ *,KTUBE
        WRITE(*,*)"Tube thickness (inches):"
        READ *,TTUBE
        WRITE(*,*)"Water side surface area of one tube (sq. in.):"
        READ *,ATUBE
        WRITE(55,6)KFIN,LFIN,WFIN,TFIN,ASAIR,ASWTR,ASFASAIR,
     +  KTUBE,TTUBE,ATUBE,RFAIR,RFWTR
      ENDIF
      IF(FLAG .EQ. 2)THEN
        WRITE(*,*)"Conduction coefficient - tube (BTU/hr ft R):"
        READ *,KTUBE
        WRITE(*,*)"Length of one tube (inches):"
        READ *,LTUBE
        WRITE(*,*)"Tube outer diameter (inches):"
        READ *,ODTUBE
        WRITE(*,*)"Tube inner diameter (inches):"
        READ *,IDTUBE
        WRITE(55,7)KFIN,LFIN,WFIN,TFIN,ASAIR,ASWTR,ASFASAIR,
     +  KTUBE,LTUBE,ODTUBE,IDTUBE,RFAIR,RFWTR
      ENDIF
C
C     Begin DO loop to compute parameters for each air side convection
C     coefficient input.
C
      DO 22 L = 1, N, 1
C
C       Using relations defined in Section IV-B-2-c, compute single fin
C       efficiency and overall efficiency.
```



```
C
        P = ((2.0*WFIN)+(2.0*TFIN))/12.0
        AX = (WFIN*TFIN)/144.0
        M = SQRT((HAIR(L)*P)/(KFIN*AX))
        LC = (LFIN+(TFIN/2.0))/12.0
        ETAF = (TANH(M*LC))/(M*LC)
        ETAO = 1.0-ASFASAIR*(1.0-ETAF)
C
C       Using relations defined in Section IV-B-3, compute overall heat
C       transfer coefficient.
C
        IF (FLAG .EQ. 1)THEN
          UAIR = 1.0/
     +    (((1.0/(ETAO*HAIR(L)*(ASAIR/144.0)))+
     +    (RFAIR/(ETAO*(ASAIR/144.0)))+
     +    (RFWTR/(ASWTR/144.0))+
     +    ((TTUBE/12.0)/(KTUBE*(ATUBE/144.0))))
     +    *(ASAIR/144.0))
        ENDIF
        IF (FLAG .EQ. 2)THEN
          UAIR = 1.0/
     +    (((1.0/(ETAO*HAIR(L)*(ASAIR/144.0)))+
     +    (RFAIR/(ETAO*(ASAIR/144.0)))+
     +    (RFWTR/(ASWTR/144.0))+
     +    (LOG(ODTUBE/IDTUBE)/(2.0*PI*(LTUBE/12.0)*KTUBE)))
     +    *(ASAIR/144.0))
        ENDIF
C
C       Print results to output file.
C
        WRITE(55,8)HAIR(L),ETAF,ETAO,UAIR
C
C     Continue DO loop to completion.
C
   22 CONTINUE
C
    4 FORMAT(//,"**HEAT EXCHANGER MODEL - OVERALL COEFFICIENT**",//,
     +"Enter the following parameters in the units",/,
     +"indicated:",/)
    5 FORMAT(" #",I2," Convection coefficient - air side",
     +" (BTU/hr sqft R):")
    6 FORMAT(////,
     +T25,"HEAT EXCHANGER OVERALL COEFFICIENT",/,
     +T25,"==================================",////,
     +T5,"Heat exchanger number:",////,
     +T8,"H-STM      = VERY LARGE",/,
     +T8,"KFIN       = ",F5.1," BTU/hr ft R",/,
     +T8,"LFIN       = ",F5.3," inches",/,
     +T8,"WFIN       = ",F5.3," inches",/
     +T8,"TFIN       = ",F6.4," inches",/,
     +T8,"ASAIR      = ",F7.1," square inches",/,
     +T8,"ASWTR      = ",F6.1," square inches",/,
     +T8,"ASF/ASAIR = ",F4.2,/,
     +T8,"KTUBE      = ",F5.1," BTU/hr ft R",/,
     +T8,"TTUBE      = ",F5.3," inches",/,
     +T8,"ATUBE      = ",F5.1," square inches",/,
     +T8,"RFAIR      = ",F6.4," hr sqft R/BTU",/,
     +T8,"RFWTR      = ",F6.4," hr sqft R/BTU",///,
     +T5,"     H-AIR      ETA 1 FIN    ETA OVERALL      U-AIR",/,
     +T5,"(BTU/hr sqft R)                              (BTU/hr sqft R)",/,
     +T5,"-----------------------------------------------------------")
    7 FORMAT(////,
     +T25,"HEAT EXCHANGER OVERALL COEFFICIENT",/,
     +T25,"==================================",////,
     +T5,"Heat exchanger number:",////,
     +T8,"H-STM      = VERY LARGE",/,
     +T8,"KFIN       = ",F5.1," BTU/hr ft R",/,
     +T8,"LFIN       = ",F5.3," inches",/,
     +T8,"WFIN       = ",F5.3," inches",/
     +T8,"TFIN       = ",F6.4," inches",/,
     +T8,"ASAIR      = ",F7.1," square inches",/,
     +T8,"ASWTR      = ",F6.1," square inches",/,
     +T8,"ASF/ASAIR = ",F4.2,/,
```

```
     +T8,"KTUBE      = ",F5.1," BTU/hr ft R",/,
     +T8,"LTUBE      = ",F5.1," inches",/,
     +T8,"ODTUBE     = ",F5.3," inches",/,
     +T8,"IDTUBE     = ",F5.3," inches",/,
     +T8,"RFAIR      = ",F6.4," hr sqft R/BTU",/,
     +T8,"RFWTR      = ",F6.4," hr sqft R/BTU",///,
     +T5,"     H-AIR      ETA 1 FIN    ETA OVERALL      U-AIR",/,
     +T5,"(BTU/hr sqft R)                            (BTU/hr sqft R)",/,
     +T5,"----------------------------------------------------------")
    8 FORMAT(T10,F4.1,T23,F4.2,T37,F4.2,T51,F4.1)
      END
```

HEAT EXCHANGER OVERALL COEFFICIENT
=================================

Heat exchanger number: B-25 (2-73-1 and 2-80-1)

```
H-STM      = VERY LARGE
KFIN       =  26.0 BTU/hr ft R
LFIN       = 0.460 inches
WFIN       = 3.510 inches
TFIN       = 0.0180 inches
ASAIR      =  3013.0 square inches
ASWTR      =  465.0 square inches
ASF/ASAIR  = 0.85
KTUBE      =  26.0 BTU/hr ft R
TTUBE      = 0.060 inches
ATUBE      =  67.0 square inches
RFAIR      = 0.0020 hr sqft R/BTU
RFWTR      = 0.0005 hr sqft R/BTU
```

H-AIR (BTU/hr sqft R)	ETA 1 FIN	ETA OVERALL	U-AIR (BTU/hr sqft R)
6.0	0.87	0.89	5.0
7.0	0.85	0.87	5.6
7.9	0.83	0.86	6.2
8.3	0.83	0.85	6.4
8.7	0.82	0.85	6.7

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HEAT EXCHANGER OVERALL COEFFICIENT

Heat exchanger number: B-70 (2-8-0, 2-40-1, 2-40-2, 2-60-1, 2-60-2)

```
H-STM     = VERY LARGE
KFIN      =  26.0 BTU/hr ft R
LFIN      = 0.460 inches
WFIN      = 3.510 inches
TFIN      = 0.0180 inches
ASAIR     =  5198.0 square inches
ASWTR     =  807.0 square inches
ASF/ASAIR = 0.85
KTUBE     =  26.0 BTU/hr ft R
TTUBE     = 0.060 inches
ATUBE     =  90.0 square inches
RFAIR     = 0.0020 hr sqft R/BTU
RFWTR     = 0.0005 hr sqft R/BTU
```

H-AIR (BTU/hr sqft R)	ETA 1 FIN	ETA OVERALL	U-AIR (BTU/hr sqft R)
9.0	0.82	0.84	6.7
9.6	0.81	0.84	7.1
9.8	0.80	0.83	7.2
10.1	0.80	0.83	7.3
10.4	0.79	0.83	7.5
10.6	0.79	0.82	7.6

```
                 HEAT EXCHANGER OVERALL COEFFICIENT
                 ==================================


Heat exchanger number:  01-24-1


   H-STM      = VERY LARGE
   KFIN       = 118.0 BTU/hr ft R
   LFIN       = 0.450 inches
   WFIN       = 1.880 inches
   TFIN       = 0.0080 inches
   ASAIR      = 17700.0 square inches
   ASWTR      = 1326.0 square inches
   ASF/ASAIR  = 0.91
   KTUBE      = 221.0 BTU/hr ft R
   LTUBE      =  34.0 inches
   ODTUBE     = 1.125 inches
   IDTUBE     = 1.035 inches
   RFAIR      = 0.0020 hr sqft R/BTU
   RFWTR      = 0.0005 hr sqft R/BTU
```

H-AIR (BTU/hr sqft R)	ETA 1 FIN	ETA OVERALL	U-AIR (BTU/hr sqft R)
7.0	0.92	0.93	6.1
7.7	0.92	0.92	6.6
8.5	0.91	0.92	7.1
9.0	0.90	0.91	7.5
9.6	0.90	0.91	7.9
9.7	0.90	0.91	8.0

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HEAT EXCHANGER OVERALL COEFFICIENT

Heat exchanger number: 01-25-1

```
H-STM     = VERY LARGE
KFIN      = 118.0 BTU/hr ft R
LFIN      = 0.438 inches
WFIN      = 1.299 inches
TFIN      = 0.0065 inches
ASAIR     =  7959.0 square inches
ASWTR     =  335.0 square inches
ASF/ASAIR = 0.96
KTUBE     = 221.0 BTU/hr ft R
LTUBE     =  24.0 inches
ODTUBE    = 0.625 inches
IDTUBE    = 0.555 inches
RFAIR     = 0.0020 hr sqft R/BTU
RFWTR     = 0.0005 hr sqft R/BTU
```

H-AIR (BTU/hr sqft R)	ETA 1 FIN	ETA OVERALL	U-AIR (BTU/hr sqft R)
8.2	0.90	0.90	6.6
8.9	0.89	0.89	7.0
9.2	0.89	0.89	7.2
10.0	0.88	0.88	7.7
10.7	0.87	0.88	8.1
11.2	0.87	0.87	8.4

HEAT EXCHANGER OVERALL COEFFICIENT

Heat exchanger number: 01-50-0

```
H-STM      = VERY LARGE
KFIN       = 118.0 BTU/hr ft R
LFIN       = 0.438 inches
WFIN       = 1.299 inches
TFIN       = 0.0065 inches
ASAIR      =  5969.0 square inches
ASWTR      =  251.0 square inches
ASF/ASAIR  = 0.96
KTUBE      = 221.0 BTU/hr ft R
LTUBE      =  18.0 inches
ODTUBE     = 0.625 inches
IDTUBE     = 0.555 inches
RFAIR      = 0.0020 hr sqft R/BTU
RFWTR      = 0.0005 hr sqft R/BTU
```

H-AIR (BTU/hr sqft R)	ETA 1 FIN	ETA OVERALL	U-AIR (BTU/hr sqft R)
7.0	0.91	0.91	5.8
7.7	0.90	0.91	6.3
8.2	0.90	0.90	6.6
8.6	0.89	0.90	6.9
9.0	0.89	0.89	7.1
9.5	0.88	0.89	7.4

HEAT EXCHANGER OVERALL COEFFICIENT

Heat exchanger number: 2-25-1

```
H-STM     = VERY LARGE
KFIN      = 118.0 BTU/hr ft R
LFIN      = 0.438 inches
WFIN      = 1.299 inches
TFIN      = 0.0100 inches
ASAIR     =  2868.0 square inches
ASWTR     =  230.0 square inches
ASF/ASAIR = 0.92
KTUBE     = 221.0 BTU/hr ft R
LTUBE     =  16.5 inches
ODTUBE    = 0.625 inches
IDTUBE    = 0.555 inches
RFAIR     = 0.0020 hr sqft R/BTU
RFWTR     = 0.0005 hr sqft R/BTU
```

H-AIR (BTU/hr sqft R)	ETA 1 FIN	ETA OVERALL	U-AIR (BTU/hr sqft R)
10.1	0.92	0.92	8.5
11.1	0.91	0.92	9.3
12.1	0.90	0.91	9.9
12.9	0.89	0.90	10.5
13.8	0.89	0.90	11.1
14.5	0.88	0.89	11.5

HEAT EXCHANGER OVERALL COEFFICIENT

Heat exchanger number: 2-16-1

```
H-STM      = VERY LARGE
KFIN       = 118.0 BTU/hr ft R
LFIN       = 0.438 inches
WFIN       = 1.299 inches
TFIN       = 0.0065 inches
ASAIR      =  3979.0 square inches
ASWTR      =  167.0 square inches
ASF/ASAIR  = 0.96
KTUBE      = 221.0 BTU/hr ft R
LTUBE      =  12.0 inches
ODTUBE     = 0.625 inches
IDTUBE     = 0.555 inches
RFAIR      = 0.0020 hr sqft R/BTU
RFWTR      = 0.0005 hr sqft R/BTU
```

H-AIR (BTU/hr sqft R)	ETA 1 FIN	ETA OVERALL	U-AIR (BTU/hr sqft R)
5.4	0.93	0.93	4.7
6.2	0.92	0.92	5.2
7.0	0.91	0.91	5.8
7.4	0.91	0.91	6.1
7.8	0.90	0.91	6.3
8.1	0.90	0.90	6.5

```
C
C                                                    Jim Hurley
C                                                    Naval Postgraduate School
C                                                    Spring 1996
C
      PROGRAM HEAT EXCHANGER MODEL NTU ANALYSIS
C
      REAL T3,T1,VDOTAIR,UAIR,ASAIR,QDOT,MDOTAIR,DAIR,T,PATM,R,
     +CAIR,CPAIR,MDOTWTRI,MDOTWTR,CWTR,CPWTR,QDOTMAX,CMIN,CMAX,CR,
     +EFF,NTUI,EFFI,NTU,ASAIRI,VDOTWTR,T2,T4,QDOTFINAL
      INTEGER FLAG
      DATA CPAIR,CPWTR,TSTD/0.24,1.0025,70.0/
C
C     Input water and air inlet temperatures, overall coefficient, air flow
C     rate, air side surface area, and heat transfer rate.
C
      OPEN(55, FILE = 'RESULTS.HXNTU')
      WRITE(*,4)
      WRITE(*,*)"Water entering temperature (degrees F):"
      READ *,T3
      WRITE(*,*)"Air stream entering temperature (degrees F):"
      READ *,T1
      WRITE(*,*)"Air stream volumetric flow rate (CFM):"
      READ *,VDOTAIR
      WRITE(*,*)"Overall heat exchanger coefficient (BTU/hr sqft R):"
      READ *,UAIR
      WRITE(*,*)"Total air side surface area (sq. in.):"
      READ *,ASAIR
      WRITE(*,*)"Heat transfer rate required (BTU/hr):"
      READ *,QDOT
C
C     Compute air mass flow rate and heat capacity rate.
C
      MDOTAIR = DAIR(TSTD)*VDOTAIR*60.0
      CAIR = MDOTAIR*CPAIR
      WRITE(55,5)T3,T1,VDOTAIR,UAIR,ASAIR,QDOT,MDOTAIR,CAIR
C
      MDOTWTRI = QDOT/(CPWTR*(T3-T1))
C
C     Begin DO loop to compute Effectiveness-NTU parameters at water mass
C     flow rates iterated up to 4876 lbm/hr (10 gpm).
C
   19 DO 23 MDOTWTR = MDOTWTRI, 4876.0, 10.0
        IF(MDOTWTR .GT. 4866.0)THEN
          WRITE(55,8)
          GO TO 99
        ENDIF
C
C     Compute water heat capacity rate, CWTR, and compare with air heat capacity
C     rate, CAIR.  If CAIR is less than CWTR, proceed with CMIN equal to CAIR.
C
        CWTR = MDOTWTR*CPWTR
        IF(CAIR .LT. CWTR)THEN
          CMIN = CAIR
          CMAX = CWTR
          FLAG = 1
          QDOTMAX = CMIN*(T3-T1)
          CR = CMIN/CMAX
          EFF = QDOT/QDOTMAX
          DO 20 NTUI = 0.0, 5.5, 0.01
            EFFI = 1.0 - EXP((1.0/CR)*(NTUI**0.22)*
     +      (EXP(-CR*(NTUI**0.78))-1.0))
            IF(ABS(EFFI-EFF) .LT. 0.01)THEN
              NTU = NTUI
              T2 = T1 + (QDOT/CAIR)
              T4 = T3 - (QDOT/CWTR)
C
C     After computations of heat capacity ratio, CR, effectiveness, EFF,
C     NTU, T2, and T4, proceed to line 22.
C
              GO TO 22
            ENDIF
   20     CONTINUE
```



```
        ENDIF
C
C     If CWTR is less than CAIR, proceed with CMIN equal to CWTR.
C
        IF(CWTR .LT. CAIR)THEN
          CMIN = CWTR
          CMAX = CAIR
          FLAG = 2
          QDOTMAX = CMIN*(T3-T1)
          CR = CMIN/CMAX
          EFF = QDOT/QDOTMAX
          DO 21 NTUI = 0.0, 5.5, 0.01
            EFFI = 1.0 - EXP((1.0/CR)*(NTUI**0.22)*
     +      (EXP(-CR*(NTUI**0.78))-1.0))
            IF(ABS(EFFI-EFF) .LT. 0.01)THEN
              NTU = NTUI
              T2 = T1 + (QDOT/CAIR)
              T4 = T3 - (QDOT/CWTR)
C
C     After computations of heat capacity ratio, CR, effectiveness, EFF,
C     NTU, T2, and T4, proceed to line 22.
C
              GO TO 22
            ENDIF
   21     CONTINUE
        ENDIF
C
C     Compute heat exchanger air side surface area and compare with actual
C     air side surface area. If computed value is less than actual value,
C     print results to output file.
C
   22   IF(NTU .LT. 1.0E-20)NTU=5.0
        ASAIRI = ((NTU*CMIN)/UAIR)*144.0
        VDOTWTR = (MDOTWTR*7.481)/(60.8*60.0)
        QDOTFINAL = EFF*CMIN*(T3-T1)
        WRITE(55,6)MDOTWTR,VDOTWTR,CMIN,CMAX,CR,EFF,NTU,ASAIRI
        IF(ASAIRI .LT. ASAIR)THEN
          WRITE(55,7)MDOTWTR,VDOTWTR,CMIN,CMAX,CR,EFF,NTU,ASAIRI,
     +    T2,T4,QDOTFINAL
          IF(FLAG .EQ. 2)THEN
            MDOTWTRI = CAIR/CPWTR + 1.0
            GO TO 19
          ENDIF
          IF(FLAG .EQ. 1)GO TO 99
        ENDIF
C
C     If computed air side surface area is greater than actual, continue
C     DO loop until solution is found or end program at water mass flow rate
C     of 4,876 lbm/hr (10 gpm).
C
   23 CONTINUE
C
    4 FORMAT(//,"****HEAT EXCHANGER MODEL - NTU ANALYSIS****",//,
     +"Enter the following parameters in the units",/,
     +"indicated:",/)
    5 FORMAT(////,
     +T25,"HEAT EXCHANGER MODEL - NTU ANALYSIS",/,
     +T25,"===================================",////,
     +T5,"Heat exchanger number:",////,
     +T8,"T3      = ",F5.1," degrees F",/,
     +T8,"T1      = ",F5.1," degrees F",/,
     +T8,"VDOTAIR = ",F6.1," CFM",/,
     +T8,"UAIR    = ",F5.1," BTU/hr sqft R",/,
     +T8,"ASAIR   = ",F7.1," square inches",/,
     +T8,"QDOT    = ",F8.1," BTU/hr",/,
     +T8,"MDOTAIR = ",F7.1," lbm/hr",/,
     +T8,"CAIR    = ",F6.1," BTU/hr R",/////,
     +T1,"  Mass   Volume     C        C       CR     Eff  NTU  Hx Surface",/,
     +T1,"  Flow    Flow     min      max                         Area",/,
     +T1,"(lbm/hr) (gpm)    (BTU/hr R)                          (sq. in.)",/,
     +T1,"------------------------------------------------------------------")
    6 FORMAT(/,T2,F6.0,T10,F5.2,T16,F6.1,T24,F7.1,T33,F4.2,T39,F4.2,
     +T45,F4.2,T52,F6.0)
```

```
    7  FORMAT(//,
      +T1,"!!!!!SOLUTION!!!!!SOLUTION!!!!!!SOLUTION!!!!!SOLUTION!!!!",//,
      +T1,"  Mass  Volume    C       C       CR    Eff  NTU  Hx Surface",/,
      +T1,"  Flow   Flow    min    max                          Area",/,
      +T1,"(lbm/hr) (gpm)   (BTU/hr R)                       (sq. in.)",/,
      +T1,"-------------------------------------------------------------",/,
      +T2,F6.0,T10,F4.2,T16,F6.1,T24,F7.1,T33,F4.2,T39,F4.2,
      +T45,F4.2,T52,F6.0,//,
      +T3,"T2          = ",F5.1," degrees F",/,
      +T3,"T4          = ",F5.1," degrees F",/,
      +T3,"QDOTFINAL = ",F8.1," BTU/hr",//,
      +T1,"!!!!!SOLUTION!!!!!SOLUTION!!!!!!SOLUTION!!!!!SOLUTION!!!!",
      +/////)
    8  FORMAT(//,T5,"At water flow rates of up to 100 gpm,",/,
      +T5,"there is no solution found.")
   99  END
C
C      ==============================================================
C
C      FUNCTION subroutines to compute temperature dependent transport
C      property of air.
C
       FUNCTION DAIR(T)
       REAL DAIR,T,PATM,R
       DATA PATM,R/14.7,53.34/
       DAIR = (PATM*144.0)/(R*(T+459.67))
       RETURN
       END
```

HEAT EXCHANGER MODEL - NTU OPTIMIZATION ANALYSIS

Heat exchanger number: 01-24-1

```
T3        = 190.0 degrees F
T1        =  62.0 degrees F
VDOTAIR   = 3680.0 CFM
UAIR      =    4.8 BTU/hr sqft R
ASAIR     = 17700.0 square inches
QDOT      =  51600.0 BTU/hr
MDOTAIR   = 16543.2 lbm/hr
CAIR      = 3970.4 BTU/hr R
```

Mass Flow (lbm/hr)	Volume Flow (gpm)	C min (BTU/hr R)	C max (BTU/hr R)	CR	Eff	NTU	Hx Surface Area (sq. in.)
402.	0.82	403.1	3970.4	0.10	1.00	5.00	60469.
412.	0.85	413.2	3970.4	0.10	0.98	3.91	48462.
422.	0.87	423.2	3970.4	0.11	0.95	3.26	41386.
432.	0.89	433.2	3970.4	0.11	0.93	2.86	37169.
442.	0.91	443.2	3970.4	0.11	0.91	2.58	34306.
452.	0.93	453.2	3970.4	0.11	0.89	2.37	32226.
462.	0.95	463.3	3970.4	0.12	0.87	2.19	30437.
472.	0.97	473.3	3970.4	0.12	0.85	2.05	29108.
482.	0.99	483.3	3970.4	0.12	0.83	1.92	27839.
492.	1.01	493.4	3970.4	0.12	0.82	1.82	26937.
502.	1.03	503.4	3970.4	0.13	0.80	1.73	26125.
512.	1.05	513.4	3970.4	0.13	0.79	1.64	25259.
522.	1.07	523.4	3970.4	0.13	0.77	1.57	24653.
532.	1.09	533.5	3970.4	0.13	0.76	1.50	24005.
542.	1.11	543.5	3970.4	0.14	0.74	1.44	23478.
552.	1.13	553.5	3970.4	0.14	0.73	1.39	23081.
562.	1.15	563.5	3970.4	0.14	0.72	1.34	22654.
572.	1.17	573.6	3970.4	0.14	0.70	1.29	22196.
582.	1.19	583.6	3970.4	0.15	0.69	1.25	21884.
592.	1.21	593.6	3970.4	0.15	0.68	1.21	21548.

602.	1.23	603.6	3970.4	0.15	0.67	1.17	21187.
612.	1.26	613.7	3970.4	0.15	0.66	1.14	20987.
622.	1.28	623.7	3970.4	0.16	0.65	1.10	20581.
632.	1.30	633.7	3970.4	0.16	0.64	1.07	20342.
642.	1.32	643.7	3970.4	0.16	0.63	1.05	20277.
652.	1.34	653.7	3970.4	0.16	0.62	1.02	20005.
662.	1.36	663.8	3970.4	0.17	0.61	0.99	19714.
672.	1.38	673.8	3970.4	0.17	0.60	0.97	19608.
682.	1.40	683.8	3970.4	0.17	0.59	0.94	19284.
692.	1.42	693.9	3970.4	0.17	0.58	0.92	19150.
702.	1.44	703.9	3970.4	0.18	0.57	0.90	19005.
712.	1.46	713.9	3970.4	0.18	0.56	0.88	18847.
722.	1.48	723.9	3970.4	0.18	0.56	0.86	18677.
732.	1.50	734.0	3970.4	0.18	0.55	0.84	18496.
742.	1.52	744.0	3970.4	0.19	0.54	0.83	18525.
752.	1.54	754.0	3970.4	0.19	0.53	0.81	18322.
762.	1.56	764.0	3970.4	0.19	0.53	0.79	18107.
772.	1.58	774.1	3970.4	0.19	0.52	0.78	18113.
782.	1.60	784.1	3970.4	0.20	0.51	0.76	17877.
792.	1.62	794.1	3970.4	0.20	0.51	0.75	17867.
802.	1.64	804.1	3970.4	0.20	0.50	0.74	17852.
812.	1.67	814.2	3970.4	0.21	0.50	0.72	17586.

!!!!!SOLUTION!!!!!SOLUTION!!!!!SOLUTION!!!!!SOLUTION!!!!

Mass Flow (lbm/hr)	Volume Flow (gpm)	C min (BTU/hr R)	C max (BTU/hr R)	CR	Eff	NTU	Hx Surface Area (sq. in.)
812.	1.67	814.2	3970.4	0.21	0.50	0.72	17586.

```
T2          =  75.0 degrees F
T4          = 126.6 degrees F
QDOTFINAL   =  51600.0 BTU/hr
```

!!!!!SOLUTION!!!!!SOLUTION!!!!!SOLUTION!!!!!SOLUTION!!!!

3961.	8.12	3970.4	3971.4	1.00	0.10	0.11	13102.

!!!!!SOLUTION!!!!!SOLUTION!!!!!SOLUTION!!!!!SOLUTION!!!!

Mass Flow (lbm/hr)	Volume Flow (gpm)	C min (BTU/hr R)	C max (BTU/hr R)	CR	Eff	NTU	Hx Surface Area (sq. in.)

```
  3961.  8.12  3970.4   3971.4  1.00  0.10  0.11   13102.

  T2        =  75.0 degrees F
  T4        = 177.0 degrees F
  QDOTFINAL =  51600.0 BTU/hr

!!!!!SOLUTION!!!!!SOLUTION!!!!!!SOLUTION!!!!!SOLUTION!!!!
```

HEAT EXCHANGER MODEL - NTU OPTIMIZATION ANALYSIS

Heat exchanger number: 2-8-0

```
T3      = 190.0 degrees F
T1      =  60.0 degrees F
VDOTAIR = 1505.0 CFM
UAIR    =   3.7 BTU/hr sqft R
ASAIR   = 5198.0 square inches
QDOT    =   5294.0 BTU/hr
MDOTAIR =  6765.6 lbm/hr
CAIR    = 1623.8 BTU/hr R
```

Mass Flow (lbm/hr)	Volume Flow (gpm)	C min (BTU/hr R)	C max (BTU/hr R)	CR	Eff	NTU	Hx Surface Area (sq. in.)
41.	0.08	40.7	1623.8	0.03	1.00	4.81	7623.
51.	0.10	50.7	1623.8	0.03	0.80	1.61	3180.

!!!!!SOLUTION!!!!!SOLUTION!!!!!!SOLUTION!!!!!SOLUTION!!!!

Mass Flow (lbm/hr)	Volume Flow (gpm)	C min (BTU/hr R)	C max (BTU/hr R)	CR	Eff	NTU	Hx Surface Area (sq. in.)
51.	0.10	50.7	1623.8	0.03	0.80	1.61	3180.

```
T2        =  63.3 degrees F
T4        =  85.7 degrees F
QDOTFINAL =   5294.0 BTU/hr
```

!!!!!SOLUTION!!!!!SOLUTION!!!!!!SOLUTION!!!!!SOLUTION!!!!

1621.	3.32	1623.8	1624.8	1.00	0.03	0.02	1264.

!!!!!SOLUTION!!!!!SOLUTION!!!!!!SOLUTION!!!!!SOLUTION!!!!

Mass Flow (lbm/hr)	Volume Flow (gpm)	C min (BTU/hr R)	C max (BTU/hr R)	CR	Eff	NTU	Hx Surface Area (sq. in.)
1621.	3.32	1623.8	1624.8	1.00	0.03	0.02	1264.

```
T2        =  63.3 degrees F
T4        = 186.7 degrees F
QDOTFINAL =   5294.0 BTU/hr
```

!!!!!SOLUTION!!!!!SOLUTION!!!!!!SOLUTION!!!!!SOLUTION!!!!

HEAT EXCHANGER MODEL - NTU OPTIMIZATION ANALYSIS

Heat exchanger number: 2-80-1

```
T3       = 190.0 degrees F
T1       =  60.0 degrees F
VDOTAIR  = 400.0 CFM
UAIR     =   4.2 BTU/hr sqft R
ASAIR    = 3013.0 square inches
QDOT     =  7747.0 BTU/hr
MDOTAIR  =  1798.2 lbm/hr
CAIR     =  431.6 BTU/hr R
```

Mass Flow (lbm/hr)	Volume Flow (gpm)	C min (BTU/hr R)	C max (BTU/hr R)	CR	Eff	NTU	Hx Surface Area (sq. in.)
59.	0.12	59.6	431.6	0.14	1.00	5.00	10216.
69.	0.14	69.6	431.6	0.16	0.86	2.16	5156.
79.	0.16	79.6	431.6	0.18	0.75	1.52	4150.
89.	0.18	89.7	431.6	0.21	0.66	1.20	3689.
99.	0.20	99.7	431.6	0.23	0.60	1.00	3418.
109.	0.22	109.7	431.6	0.25	0.54	0.86	3235.
119.	0.24	119.7	431.6	0.28	0.50	0.75	3079.
129.	0.27	129.8	431.6	0.30	0.46	0.67	2981.

!!!!!SOLUTION!!!!!SOLUTION!!!!!!SOLUTION!!!!!SOLUTION!!!!

Mass Flow (lbm/hr)	Volume Flow (gpm)	C min (BTU/hr R)	C max (BTU/hr R)	CR	Eff	NTU	Hx Surface Area (sq. in.)
129.	0.27	129.8	431.6	0.30	0.46	0.67	2981.

```
T2        =  78.0 degrees F
T4        = 130.3 degrees F
QDOTFINAL =  7747.0 BTU/hr
```

!!!!!SOLUTION!!!!!SOLUTION!!!!!!SOLUTION!!!!!SOLUTION!!!!

431.	0.88	431.6	432.6	1.00	0.14	0.16	2367.

!!!!!SOLUTION!!!!!SOLUTION!!!!!!SOLUTION!!!!!SOLUTION!!!!

Mass Flow (lbm/hr)	Volume Flow (gpm)	C min (BTU/hr R)	C max (BTU/hr R)	CR	Eff	NTU	Hx Surface Area (sq. in.)
431.	0.88	431.6	432.6	1.00	0.14	0.16	2367.

```
T2        =  78.0 degrees F
T4        = 172.1 degrees F
QDOTFINAL =  7747.0 BTU/hr
```

!!!!!SOLUTION!!!!!SOLUTION!!!!!!SOLUTION!!!!!SOLUTION!!!!

APPENDIX D. EFFECTIVENESS - NTU ANALYSIS TABULAR RESULTS

UNIT HEATER B-25 (2-73-1, 2-80-1)																			
HYDRONIC																			
COIL DIMENSIONS					AIR PROPERTIES						WATER PROPERTIES								
AFF	ASAIR	DH		RHO	CP	MU	K	PR	T1		CP	T3							
62	3013	0.38		0.0749	0.24	1.23E-05	0.0148	0.72	60		1.002	190							
CFM	MDOTAIR	G		RE	JH	H-AIR	H-WTR	UAIR	CAIR		MDOTWT	CWTR	QMAX	NTU	CR	EPS	Q	T2	T4
400	1798	1.16		2981	0.0048	6.0	4053	4.9	431		4876	4886	56085	0.24	0.09	0.21	11713	87	188
500	2247	1.45		3726	0.0045	7.0	4053	5.6	539		4876	4886	70106	0.22	0.11	0.19	13487	85	187
600	2696	1.74		4471	0.0042	7.9	4053	6.1	647		4876	4886	84128	0.20	0.13	0.18	14807	83	187
700	3146	2.03		5217	0.0038	8.3	4053	6.4	755		4876	4886	98149	0.18	0.15	0.16	15663	81	187
800	3595	2.32		5962	0.0035	8.7	4053	6.6	863		4876	4886	112170	0.16	0.18	0.15	16270	79	187
HYDRONIC WITH VARIATION OF T3																			
T3																			
205																			
CFM	MDOTAIR	G		RE	JH	H-AIR	H-WTR	UAIR	CAIR		MDOTWT	CWTR	QMAX	NTU	CR	EPS	Q	T2	T4
400	1798	1.16		2981	0.0048	6.0	4053	4.9	431		4876	4886	62556	0.24	0.09	0.21	13064	90	202
STEAM																			
COIL DIMENSIONS						AIR PROPERTIES					STEAM PROPERTIES								
AFF	ASAIR	DH		RHO	CP	MU	K	PR	T1		CP	T3	HFG						
62	3013	0.38		0.0749	0.24	1.23E-05	0.0148	0.72	60		INF.	240	952						
CFM	MDOTAIR	G		RE	JH	H-AIR	H-STM	UAIR	CAIR		MDOTST	CSTM	QMAX	NTU	CR	EPS	Q	T2	
400	1798	1.16		2981	0.0048	6.0	>>0	5.0	431		17.6	INF.	77656	0.24	0.00	0.22	16722	99	

UNIT HEATER B-70 (2-8-0, 2-40-1, 2-40-2, 2-60-1, 2-60-2)																			
HYDRONIC																			
COIL DIMENSIONS					AIR PROPERTIES						WATER PROPERTIES								
AFF	ASAIR	DH		RHO	CP	MU	K	PR	T1		CP	T3							
120	5198	0.38		0.0749	0.24	1.23E-05	0.0148	0.72	60		1.002	190							
CFM	MDOTAIR	G		RE	JH	H-AIR	H-WTR	UAIR	CAIR		MDOTWT	CWTR	QMAX	NTU	CR	EPS	Q	T2	T4
1505	6763	2.25		5795	0.0037	9.0	4053	6.7	1623		4876	4886	211020	0.15	0.33	0.13	28214	77	184
1800	8089	2.70		6931	0.0033	9.6	4053	7.0	1941		4876	4886	252383	0.13	0.40	0.12	29661	75	184
2100	9437	3.15		8086	0.0029	9.8	4053	7.1	2265		4876	4886	294447	0.11	0.46	0.10	30273	73	184
2400	10786	3.60		9241	0.0026	10.1	4053	7.2	2589		4876	4886	336511	0.10	0.53	0.09	30834	72	184
2700	12134	4.04		10396	0.0024	10.4	4053	7.4	2912		4876	4886	378575	0.09	0.60	0.08	31757	71	184
3000	13482	4.49		11551	0.0022	10.6	4053	7.5	3236		4876	4886	420638	0.08	0.66	0.08	32260	70	183
HYDRONIC WITH VARIATION OF T3																			
T3																			
205																			
CFM	MDOTAIR	G		RE	JH	H-AIR	H-WTR	UAIR	CAIR		MDOTWT	CWTR	QMAX	NTU	CR	EPS	Q	T2	T4
1505	6763	2.25		5795	0.0037	9.0	4053	6.7	1623		4876	4886	235369	0.15	0.33	0.13	31469	79	199
STEAM																			
COIL DIMENSIONS						AIR PROPERTIES					STEAM PROPERTIES								
AFF	ASAIR	DH		RHO	CP	MU	K	PR	T1		CP	T3	HFG						
120	5198	0.38		0.0749	0.24	1.23E-05	0.0148	0.72	60		INF.	240	952						
CFM	MDOTAIR	G		RE	JH	H-AIR	H-STM	UAIR	CAIR		MDOTST	CSTM	QMAX	NTU	CR	EPS	Q	T2	
1505	6763	2.25		5795	0.0037	9.0	>>0	6.7	1623		42.5	INF.	292182	0.15	0.00	0.14	40445	85	

DUCT HEATER 01-24-1

HYDRONIC

COIL DIMENSIONS				AIR PROPERTIES							WATER PROPERTIES								
AFF	ASAIR	DH		RHO	CP	MU	K	PR	T1		CP	T3							
405	17700	0.25		0.0749	0.24	1.23E-05	0.0148	0.72	62		1.002	190							
CFM	MDOTAIR	G		RE	JH	H-AIR	H-WTR	UAIR	CAIR		MDOTWT	CWTR	QMAX	NTU	CR	EPS	Q	T2	T4
3680	16538	1.63		2762	0.004	7.0	4053	5.7	3969		4876	4886	508045	0.18	0.81	0.15	74778	81	175
4500	20223	2.00		3378	0.0036	7.7	4053	6.2	4854		4876	4886	621251	0.16	0.99	0.13	81146	79	173
5400	24268	2.40		4053	0.0033	8.5	4053	6.7	5824		4876	4886	625376	0.17	0.84	0.14	88237	77	172
6300	28312	2.80		4729	0.003	9.0	4053	7.0	6795		4876	4886	625376	0.18	0.72	0.15	92857	76	171
7200	32357	3.20		5404	0.0028	9.6	4053	7.4	7766		4876	4886	625376	0.19	0.63	0.16	98424	75	170
7360	33076	3.27		5524	0.00275	9.7	4053	7.4	7938		4876	4886	625376	0.19	0.62	0.16	98586	74	170

HYDRONIC WITH VARIATION OF T3

T3																			
205																			
CFM	MDOTAIR	G		RE	JH	H-AIR	H-WTR	UAIR	CAIR		MDOTWT	CWTR	QMAX	NTU	CR	EPS	Q	T2	T4
3680	16538	1.63		2762	0.004	7.0	4053	5.7	3969		4876	4886	567581	0.18	0.81	0.15	83541	83	188

STEAM

COIL DIMENSIONS						AIR PROPERTIES					STEAM PROPERTIES								
AFF	ASAIR	DH		RHO	CP	MU	K	PR	T1		CP	T3	HFG						
405	17700	0.25		0.0749	0.24	1.23E-05	0.0148	0.72	62		INF.	240	952						
CFM	MDOTAIR	G		RE	JH	H-AIR	H-STM	UAIR	CAIR		MDOTST	CSTM	QMAX	NTU	CR	EPS	Q	T2	
3680	16538	1.63		2762	0.0048	7.0	>>0	6.1	3969		127.7	INF.	706500	0.19	0.00	0.17	121615	93	

DUCT HEATER 01-25-1

HYDRONIC

COIL DIMENSIONS				AIR PROPERTIES						WATER PROPERTIES	
AFF	ASAIR	DH	RHO	CP	MU	K	PR	T1		CP	T3
155	7959	0.154	0.0749	0.24	1.23E-05	0.0148	0.72	-30		1.002	190

CFM	MDOTAIR	G	RE	JH	H-AIR	H-WTR	UAIR	CAIR	MDOTWT	CWTR	QMAX	NTU	CR	EPS	Q	T2	T4
1400	6292	1.62	1691	0.0047	8.2	4053	6.4	1510	4876	4886	332196	0.23	0.31	0.20	66393	14	176
1700	7640	1.97	2054	0.0042	8.9	4053	6.8	1834	4876	4886	403381	0.20	0.38	0.18	71200	9	175
2000	8988	2.32	2416	0.0037	9.2	4053	6.9	2157	4876	4886	474566	0.18	0.44	0.15	73016	4	175
2300	10336	2.67	2779	0.0035	10.0	4053	7.4	2481	4876	4886	545751	0.16	0.51	0.14	78375	2	174
2600	11684	3.02	3141	0.0033	10.7	4053	7.8	2804	4876	4886	616936	0.15	0.57	0.13	82704	-1	173
2800	12583	3.25	3383	0.0032	11.2	4053	8.0	3020	4876	4886	664393	0.15	0.62	0.13	84919	-2	173

HYDRONIC WITH VARIATION OF T3

T3
205

CFM	MDOTAIR	G	RE	JH	H-AIR	H-WTR	UAIR	CAIR	MDOTWT	CWTR	QMAX	NTU	CR	EPS	Q	T2	T4
1400	6292	1.62	1691	0.0047	8.2	4053	6.4	1510	4876	4886	354846	0.23	0.31	0.20	70920	17	190

STEAM

COIL DIMENSIONS				AIR PROPERTIES					STEAM PROPERTIES		
AFF	ASAIR	DH	RHO	CP	MU	K	PR	T1	CP	T3	HFG
155	7959	0.154	0.0749	0.24	1.23E-05	0.0148	0.72	-30	INF.	240	952

CFM	MDOTAIR	G	RE	JH	H-AIR	H-STM	UAIR	CAIR	MDOTST	CSTM	QMAX	NTU	CR	EPS	Q	T2
1400	6292	1.62	1691	0.0047	8.2	>>0	6.6	1510	91.9	INF.	407696	0.24	0.00	0.21	87498	28

DUCT HEATER 01-50-0

HYDRONIC

COIL DIMENSIONS				AIR PROPERTIES							WATER PROPERTIES								
AFF	ASAIR	DH		RHO	CP	MU	K	PR	T1		CP	T3							
116	5969	0.154		0.0749	0.24	1.23E-05	0.0148	0.72	-30		1.002	190							
CFM	MDOTAIR	G		RE	JH	H-AIR	H-WTR	UAIR	CAIR		MDOTWT	CWTR	QMAX	NTU	CR	EPS	Q	T2	T4
750	3371	1.16		1211	0.0056	7.0	4053	5.6	809		4876	4886	177962	0.29	0.17	0.24	43215	23	181
900	4045	1.39		1453	0.0051	7.7	4053	6.1	971		4876	4886	213555	0.26	0.20	0.22	47509	19	180
1050	4719	1.63		1695	0.0047	8.2	4053	6.4	1132		4876	4886	249147	0.23	0.23	0.20	50341	14	180
1200	5393	1.86		1937	0.0043	8.6	4053	6.6	1294		4876	4886	284740	0.21	0.26	0.18	52374	10	179
1350	6067	2.09		2179	0.004	9.0	4053	6.8	1456		4876	4886	320332	0.19	0.30	0.17	54312	7	179
1500	6741	2.32		2421	0.0038	9.5	4053	7.1	1618		4876	4886	355925	0.18	0.33	0.16	56893	5	178

HYDRONIC WITH VARIATION OF T3

T3
205

CFM	MDOTAIR	G		RE	JH	H-AIR	H-WTR	UAIR	CAIR		MDOTWT	CWTR	QMAX	NTU	CR	EPS	Q	T2	T4
750	3371	1.16		1211	0.0056	7.0	4053	5.6	809		4876	4886	190096	0.29	0.17	0.24	46162	27	196

STEAM

COIL DIMENSIONS						AIR PROPERTIES					STEAM PROPERTIES								
AFF	ASAIR	DH		RHO	CP	MU	K	PR	T1		CP	T3	HFG						
116	5969	0.154		0.0749	0.24	1.23E-05	0.0148	0.72	-30		INF.	240	952						
CFM	MDOTAIR	G		RE	JH	H-AIR	H-STM	UAIR	CAIR		MDOTST	CSTM	QMAX	NTU	CR	EPS	Q	T2	
750	3371	1.16		1211	0.0056	7.0	>>0	5.8	809		59.0	INF.	218408	0.30	0.00	0.26	56155	39	

DUCT HEATER 2-25-1																			
HYDRONIC																			
COIL DIMENSIONS					AIR PROPERTIES						WATER PROPERTIES								
AFF	ASAIR	DH		RHO	CP	MU	K	PR	T1		CP	T3							
109	2868	0.312		0.0749	0.24	1.23E-05	0.0148	0.72	40		1.002	190							
CFM	MDOTAIR	G		RE	JH	H-AIR	H-WTR	UAIR	CAIR		MDOTWT	CWTR	QMAX	NTU	CR	EPS	Q	T2	T4
1050	4719	1.73		3654	0.0054	10.1	4053	8.3	1132		4876	4886	169873	0.15	0.23	0.13	22526	60	185
1250	5618	2.06		4351	0.005	11.1	4053	9.0	1348		4876	4886	202230	0.13	0.28	0.12	24515	58	185
1450	6516	2.39		5047	0.0047	12.1	4053	9.7	1564		4876	4886	234587	0.12	0.32	0.11	26473	57	185
1650	7415	2.72		5743	0.0044	12.9	4053	10.2	1780		4876	4886	266944	0.11	0.36	0.10	27905	56	184
1850	8314	3.05		6439	0.0042	13.8	4053	10.7	1995		4876	4886	299300	0.11	0.41	0.10	29316	55	184
2100	9437	3.46		7309	0.0039	14.5	4053	11.1	2265		4876	4886	339746	0.10	0.46	0.09	30485	53	184
HYDRONIC WITH VARIATION OF T3																			
T3																			
205																			
CFM	MDOTAIR	G		RE	JH	H-AIR	H-WTR	UAIR	CAIR		MDOTWT	CWTR	QMAX	NTU	CR	EPS	Q	T2	T4
1050	4719	1.73		3654	0.0054	10.1	4053	8.3	1132		4876	4886	186861	0.15	0.23	0.13	24779	62	200
STEAM																			
COIL DIMENSIONS						AIR PROPERTIES					STEAM PROPERTIES								
AFF	ASAIR	DH		RHO	CP	MU	K	PR	T1		CP	T3	HFG						
109	2868	0.312		0.0749	0.24	1.23E-05	0.0148	0.72	40		INF.	240	952						
CFM	MDOTAIR	G		RE	JH	H-AIR	H-STM	UAIR	CAIR		MDOTST	CSTM	QMAX	NTU	CR	EPS	Q	T2	
1050	4719	1.73		3654	0.0054	10.1	>>0	8.5	1132		33.0	INF.	226498	0.15	0.00	0.14	31449	68	

DUCT HEATER 2-16-1

HYDRONIC

COIL DIMENSIONS				AIR PROPERTIES						WATER PROPERTIES	
AFF	ASAIR	DH	RHO	CP	MU	K	PR	T1		CP	T3
77	3979	0.154	0.0749	0.24	1.23E-05	0.0148	0.72	45		1.002	190

CFM	MDOTAIR	G	RE	JH	H-AIR	H-WTR	UAIR	CAIR	MDOTWT	CWTR	QMAX	NTU	CR	EPS	Q	T2	T4
350	1573	0.82	851	0.0062	5.4	4053	4.5	377	4876	4886	54737	0.33	0.08	0.28	15152	85	187
425	1910	0.99	1034	0.0058	6.2	4053	5.1	458	4876	4886	66466	0.31	0.09	0.26	17313	83	186
500	2247	1.17	1216	0.0056	7.0	4053	5.6	539	4876	4886	78196	0.29	0.11	0.25	19158	81	186
575	2584	1.34	1398	0.0051	7.4	4053	5.9	620	4876	4886	89925	0.26	0.13	0.23	20385	78	186
650	2921	1.52	1581	0.0048	7.8	4053	6.1	701	4876	4886	101654	0.24	0.14	0.21	21277	75	186
700	3146	1.63	1702	0.0046	8.1	4053	6.3	755	4876	4886	109474	0.23	0.15	0.20	22057	74	185

HYDRONIC WITH VARIATION OF T3

T3
205

CFM	MDOTAIR	G	RE	JH	H-AIR	H-WTR	UAIR	CAIR	MDOTWT	CWTR	QMAX	NTU	CR	EPS	Q	T2	T4
350	1573	0.82	851	0.0062	5.4	4053	4.5	377	4876	4886	60399	0.33	0.08	0.28	16720	89	202

STEAM

COIL DIMENSIONS			AIR PROPERTIES						STEAM PROPERTIES		
AFF	ASAIR	DH	RHO	CP	MU	K	PR	T1	CP	T3	HFG
77	3979	0.154	0.0749	0.24	1.23E-05	0.0148	0.72	45	INF.	240	952

CFM	MDOTAIR	G	RE	JH	H-AIR	H-STM	UAIR	CAIR	MDOTST	CSTM	QMAX	NTU	CR	EPS	Q	T2
350	1573	0.82	851	0.0062	5.4	>>0	4.7	377	22.5	INF.	73612	0.34	0.00	0.29	21428	102

APPENDIX E. MANUFACTURER'S DATA FOR LARGER CAPACITY FANS COMPATIBLE WITH B25 UNIT HEATERS

The
New York Blower
Company®

7660 QUINCY STREET—WILLOWBROOK, ILLINOIS 60521

MODEL SN DIRECT-DRIVE SUPPLY CAPACITIES

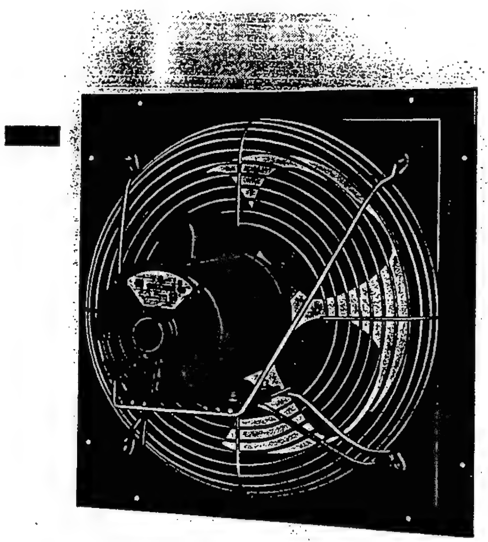
Model no.	HP	RPM	CFM						Max* BHP
			Free air	1/10" SP	1/8" SP	1/4" SP	3/8" SP	1/2" SP	
SN82-H	1/20	1550	442	316	270	—	—	—	—
SN82-H-3	†1/20	1550/1300/1100	442/371/314	316/265/224	270/226/192	—	—	—	—
SN102-H	1/20	1550	870	755	720	—	—	—	—
SN102-H-3	†1/20	1550/1300/1100	870/730/617	755/633/536	720/604/511	—	—	—	—
SN122-M	1/12	1075	1150	920	850	—	—	—	—
SN122-H	1/4	1725	1815	1675	1650	1475	—	—	—
SN122-MH	1/4	1725/1140	1815/1200	1675/1106	1650/1090	1475/975	—	—	—
SN142-M	1/12	1050	1350	1160	1100	—	—	—	—
SN142-H	1/4	1725	2100	1990	1960	1840	1680	—	—
SN142-MH	1/4	1725/1140	2100/1390	1990/1315	1960/1295	1840/1216	1680/1110	—	—
SN162-M	1/4	1140	2000	1800	1750	1450	—	—	—
SN162-H	1/2	1750	2950	2830	2800	2650	2500	—	.45
SN162-MH	1/3	1725/1140	2900/1915	2790/1840	2760/1756	2600/1718	2440/1610	—	.45
SN182-M	1/4	1140	2610	2400	2340	1960	—	—	—
SN182-H	1/2	1725	3920	3750	3700	3490	3280	3000	.57
SN182-MH	1/2	1725/1140	3920/2590	3750/2480	3700/2440	3490/2305	3280/2165	3000/1980	.57
SN202-M	1/4	1140	3570	3260	3200	2810	—	—	—
SN202-H	3/4	1725	5300	5100	5000	4820	4600	4350	.92
SN242-L	1/4	1140	4400	4150	4080	3700	—	—	—
SN242-M	1/2	1140	5380	5100	5030	4650	4200	—	.52
SN242-H	★3/4	1140	6400	6100	6020	5600	5120	4480	.79

NOTE Static pressure rating on multispeed fans is at the higher speed. Low speed capacities are shown for the indentical system.

* Maximum BHP over cataloged range. Motors are rated on internal temperature rise rather than nameplate HP.

† Shaded-pole motor. Three-speed capacities shown are obtainable with 3-speed switch furnished with unit.

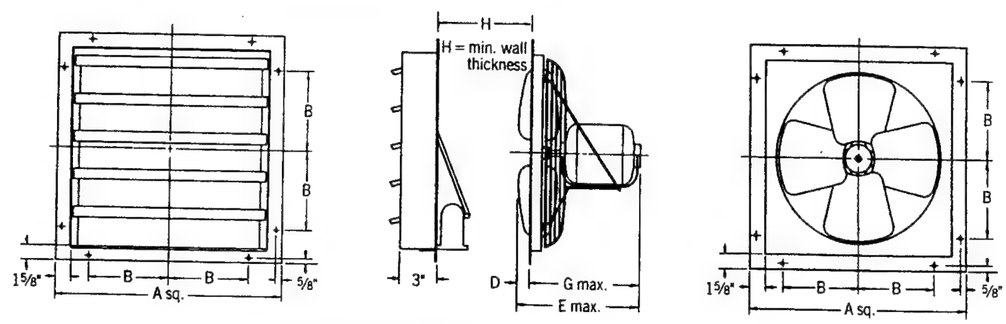
★ Available in 3-phase only.



MODEL N

DIRECT-DRIVE PROPELLER FANS
EXHAUST OR SUPPLY

- **Eight wheel diameters**—8" through 24".
- **243 to 6420 CFM**— up to ½" static pressure.
- **Panels**—square steel construction with streamlined venturi inlet...venturi is reversed in supply-fan panels...baked-green alkyd finish.
- **Wheels**—aluminum blades with steel hubs.
- **Motor mounts**—wire-guard-type motor mount [see photo at left] is standard on all direct-drive units...guard is zinc-plated steel.
- **Motors**—standard motors are totally enclosed air over with pre-lubricated ball bearings except 1/12 and 1/20 HP motors, which are shaded-pole totally enclosed permanently lubricated sleeve-bearing type.
  Motors ¼ HP and larger are suitable for either horizontal or vertical service...specify "for vertical mounting" to have wheel locked to motor shaft...1/20 and 1/12 HP motors are not suitable for vertical service.



SPECIFICATIONS DIMENSIONS IN INCHES.

Application	Model no.	Wheel diameter	A	B	D	E†	G†	H minimum Auto-matic	H minimum Motor oper.	Mounting hole no. and diameter Fan	Mounting hole no. and diameter Shutter	Weight* [lbs.]
EXHAUST	EN82-	8	13¼	3	—	10¼	10¼	1⅝	4½	8 - 5/16	8 - 9/32	25
	EN102-	10	15¼	4	¼	10⅜	10⅛	1⅝	4¾	8 - 5/16	8 - 9/32	29
	EN122-	12	17¼	5	⅞	11½	10¾	2	5⅜	8 - 5/16	8 - 9/32	35
	EN142-	14	20¼	6½	⅝	11¼	10⅝	2	5⅛	8 - 5/16	8 - 9/32	40
	EN162-	16	23¼	8	1	12	11	2	5½	8 - 5/16	8 - 9/32	50
	EN182-	18	24¼	8½	⅝	11½	10⅞	2	5⅛	8 - 5/16	8 - 9/32	65
	EN202-	20	27¼	10	⅞	12⅞	12	2	5⅜	8 - 5/16	8 - 9/32	80
	EN242-	24	30¼	11½	1	13⅜	12⅜	2	5½	8 - 5/16	8 - 9/32	95
SUPPLY	SN82-	8	13¼	3	—	10¼	10¼	Auto-matic shutter not available	9½	8 - 5/16	8 - 9/32	25
	SN102-	10	15¼	4	—	10¼	10¼		9½	8 - 5/16	8 - 9/32	29
	SN122-	12	17¼	5	—	11	11		9½	8 - 5/16	8 - 9/32	35
	SN142-	14	20¼	6½	—	11⅝	11⅝		9½	8 - 5/16	8 - 9/32	40
	SN162-	16	23¼	8	⅛	11¼	11⅛		9½	8 - 5/16	8 - 9/32	50
	SN182-	18	24¼	8½	—	12	12		9½	8 - 5/16	8 - 9/32	65
	SN202-	20	27¼	10	—	12½	12½		9½	8 - 5/16	8 - 9/32	80
	SN242-	24	30¼	11½	¼	12½	12¼		9½	8 - 5/16	8 - 9/32	95

† E and G based on longest motor used for each size fan. * Shipping weights shown are maximum and include totally enclosed motors and weight of packaging.
NOTE: Exhaust units are available with either automatic or motorized shutters. Supply units require motorized supply shutter.
When ordering, specify complete model number as shown on page 3. Dimensions not to be used for construction unless certified. Tolerance: ± ⅛"

APPENDIX F. COMPUTATION OF AIR SIDE CONVECTION COEFFICIENT USING MANUFACTURER'S DATA

Using heater 01-50-0's operating condition of 750 CFM

$$\Rightarrow V_{air} = \frac{750 \ FT^3/min}{A_{fr}} = \frac{750 \ FT^3/min}{(1 \times 1.5) \ FT^2} = 500 \ FT/min$$

$\Rightarrow$ Table 2 @ 12 FINS/inch, 1 row, 500 FT/min

$$\Delta T_{air} = 77.2 \ °F$$

$$\dot{Q} = C_{AIR} \Delta T = \dot{m} c_{p_{air}} \Delta T = \rho A_{fr} V c_{p_{air}} \Delta T$$

$$= (0.0749 \ lbm/ft^3)(1.5 \ FT^2)(500 \ FT/min)\left(0.24 \ \frac{BTU}{lbm F}\right)(77.2 \ F)(60 \ min/hr)$$

$$= 62449 \ BTU/hr$$

$$\Rightarrow \varepsilon = \frac{\dot{Q}}{\dot{Q}_{MAX}} = \frac{\dot{Q}}{C_{min}(T_s - T_1)} = \frac{\dot{Q}}{C_{AIR}(T_s - T_1)}$$

$$= \frac{62449 \ BTU/hr}{809 \ \frac{BTU}{hr F}(227 - 0) \ F} = 0.34$$

$$\Rightarrow NTU = -\ln(1 - \varepsilon)$$

$$= -\ln(1 - 0.34)$$

$$= 0.42$$

$$\Rightarrow NTU = \frac{UA}{C_{min}}$$

$$\Rightarrow U = \frac{NTU\ C_{min}}{A_s} = \frac{NTU\ C_{air}}{A_{s_{AIR}}}$$

$$= \frac{(0.42)(809\ \frac{BTU}{hr\,F})}{5969\ in^2 / (144\ in^2/ft^2)} = 8.2\ \frac{BTU}{hr\,ft^2\,F}$$

$$\Rightarrow \frac{1}{hA} = \frac{1}{\eta_o hA} \Rightarrow h_{AIR} = \frac{U}{\eta_o} \quad \frac{8.2\ \frac{BTU}{hr\,ft^2\,F}}{0.92} = 8.9\ \frac{BTU}{hr\,ft^2\,F}$$

$$\Rightarrow \dot{Q} = h_{AIR} A_{s_{AIR}} (T_s - T_1)$$

$$T_1 = T_s - Q / h_{AIR} A_{s_{AIR}}$$

$$= 227\,°F - \frac{62449\ BTU/hr}{(8.9\ \frac{BTU}{hr\,ft^2\,F})\left(\frac{5969\ in^2}{144\ in^2/ft^2}\right)}$$

$$= \underline{\underline{58\,°F}}$$

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